Copyright, 1919, by the American Society of Naval Engineers.

JOURNAL
OF THE
AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXXI. FEBRUARY, 1919. No. 1.

The Society as a body is not responsible for statements made by individual members.

COUNCIL OF THE SOCIETY
(Under whose supervision this number is published).
Rear Admiral H. P. Norton, U.S.N. Rear Admiral T. W. Kinkaid, U.S.N.
Captain Robert Stocker, (C.C.) U.S.N. Commander S. M. Robinson, U.S.N.
Commander W. W. White, U.S.N. (Retired). Capt. of Eng. M. W. Torbet, C. G.
Lieutenant Commander F. W. Sterling, U.S.N. (Retired).

# THE ACHIEVEMENTS OF NAVAL ENGINEERING IN THIS WAR.*

By Lieut. Comdr. William L. Cathcart, U. S. N., R. F., Member.

During these more than four years of war we have heard much in criticism of the Silent Fleets—of the mute guns of that vast Allied Armada waiting tensely, like a crouching lion, for the German High Seas Fleet, which, save for its half-hearted dash at Jutland, never came, until the end in a surrender so ignoble that it sickened the hearts of seamen.

And yet, notwithstanding these criticisms, the ex-Kaiser, his craven commanders on land and sea, and now all Germany know that to those Silent Fleets is due primarily the shattering of their dream of world dominion—a dream whose realization meant to us and to our allies but world despair.

This assertion of the paramount value of sea power involves no detraction from the honor fitly due those magnificent

*This address was delivered before the American Society of Mechanical Engineers and printed in an abridged form in the Journal of that society. It is here published in full with additions.

allied armies, the white crosses of whose dead crowd all Europe. And it minimizes in no way the achievement of the superb troops of the American Expeditionary Force, who, in but half a year's hard fighting against the flower of Germany's armies, have won undying glory for our flag.

But, predominating sea power was the foundation on which all of these victories on the land were based. From the beginning, the British Fleet attained the ultimate object of all sea warfare, in making impotent the enemy's naval strength in surface warships—the only kind that count in the end—and that impotence has had a vital effect on the conduct and success of the war we have waged.

The reason is clear. Every one knows now that an army on the land or a fleet on the sea becomes helpless if its lines of communication with its base are cut. In this conflict, the Western front was the decisive theater of war. And for the allied armies on that front the ultimate bases of supply and reënforcement lay beyond the sea—in Great Britain, her colonies and Dominions; in India and Algiers; and, most important of all, in distant America.

So, the absolutely vital lines of communication of those armies stretched like a vast network across the Seven Seas; and, if Germany's High Seas Fleet had been free to wreak its ruthless will, those lines would have been quickly cut, England would have been isolated, America powerless to aid, and, long before we could have entered it, the war would have ended in bitter tragedy for all mankind that is worth while.

## OUR NAVY IN THIS WAR.

While the defeat of the Central Powers is thus due primarily to the early strength and continuous growth of the British fleet, our own Navy has had a far from inconsiderable part in the decisive work of the war's last year, in its service in the Atlantic and the North, White, and Mediterranean Seas, with the ships of the allied navies. On the world's seas, wherever war clouds lowered, our flag flew.

From the very day of America's entry into the war our Navy grew by leaps and bounds, until it became 600,000 men strong and operated about 2,000 armed vessels and transports. On December 1 we had in European waters a total of 338 ships of all classes, carrying 5,000 officers and 70,000 enlisted men, all under command of Vice Admiral Sims. These vessels comprised:

First, a squadron of dreadnaughts in the North Sea; second, cruisers serving with the British in the White Sea and the North Sea, with the British and French in the Atlantic, and with the French, Italian and Japanese in the Mediterranean; third, about a hundred Destroyers operating from several European bases; fourth, a swarm of submarine chasers dashing through British, French and Italian coastal waters; fifth, a flotilla of submarines engaged in hunting enemy boats of their own kind; sixth, a number of small patrol vessels—Coast Guard ships, yachts, seagoing tugs and the like; and, finally, some naval auxiliaries, such as mine layers and repair and hospital ships.

The story of that fleet of ours in those distant, stormy, mist-enshrouded seas, is a very noble one in its record of unceasing toil, of daring and self-sacrifice, of the gallant deeds of the living and the dead. But it cannot be fully told until the censor's ban is lifted.

In his annual report for 1918 the Secretary of the Navy gives averages aggregating 626,000 miles steamed per month as an indication of the work done by all classes of U. S. Naval vessels, engaged in naval duties only, in European war areas. During April, May and June, 1918, our destroyers there escorted 121 troop-ship convoys, consisting of 775 vessels, and during the same period they gave similar guard to 171 merchant convoys, aggregating 1,763 ships.

But the story of the Navy in this war is very far from ending with the record of its ships in European waters. To meet the submarine menace, the United States Naval Cruiser

and Transport Service was organized under command of Vice Admiral Gleaves. When the armistice was granted, this force consisted of 24 cruisers and 42 transports, manned by about 3,000 officers and 42,000 enlisted men of the Navy.

Of the total number of American troops transported to Europe, 46¼ per cent were carried in American ships, and of these all but 2½ per cent were carried in U. S. Naval Transports. In addition, all troops transported in American ships were escorted by U. S. men-of-war. Admiral Gleaves's force thus carried nearly a million troops eastward across 3,000 miles of sea without the loss of a single soldier—an achievement, which, for rapidity of execution and the distance covered, is without parallel in the history of navies. During the last three months of the war, for every minute of the day and night, seven American soldiers, their equipment, and the maintenance for them, arrived in France. As to all this, Admiral Jellicoe says that it was the assistance rendered by the United States Navy which had made the convoy system possible, and that it was this system which had saved the situation.

And finally, we have maintained also a huge Atlantic Fleet, under command of Admiral Mayo, whose vessels have not only guarded our coasts and the near-by ocean lanes, but have formed a great training school from which have come the picked men for European service, the gunners for many hundreds of merchant vessels carrying armed guards, and the crews for Army transports and for the Naval Overseas Transportation Service—that is, the vessels built by the Emergency Fleet Corporation.

When we remember that the maritime area of this country forms so small a part of the whole, the wonder is that, in so brief a period, 600,000 men, coming largely from interior States, could be trained for efficient service afloat in our home and foreign fleets. Sir Eric Geddes justly terms the achievement of the Bureau of Navigation in this work as "an amazing feat."

But what is thus true of one Bureau in this war is equally true of all in the Navy Department—the great shore establishment which stands behind the Navy afloat, and without whose full efficiency effective service by our Fleet would be impossible.

So, what I shall have to tell in a moment of the record made by the Bureau of Steam Engineering should be regarded as but a sample of what all these Bureaus have done.

From the beginning of the war the entire Navy Department has been working, in a clean-cut American way, at maximum efficiency under forced draft.

## NAVAL ENGINEERING.

As to my subject this evening, "The Achievement of Naval Engineering in This War," may I say that I shall limit my address to the work of the Bureau of Steam Engineering, and shall be able to handle even that work only in a very general way, since the Bureau's field, which is dynamical engineering largely, covers all steam and internal-combustion engines for the Navy; the bulk of its electrical apparatus for surface ships, submarines and aircraft; the machinery for aircraft; the generation and supply of gas for observation balloons and dirigibles; and, finally, the design and supply of the entire wireless equipment of all shore stations for radio telegraphy in the United States, in our island possessions, and on every vessel, merchant or naval, flying the American flag.

In a broad sense, however, there is much other "naval engineering," although it is static largely, as in the fundamentally important work of the Bureau of Construction and Repair on hulls and their fittings, on stabilizers for ships, and in the Naval Aircraft Factory; and again in the field of the Bureau of Yards and Docks which covers dry docks, buildings and grounds. And, further, the Bureau of Ordnance, in its guns, is but building a special form of internal-combustion engine, and its armor is a product of statical engineering skill.

In fact, in this broad sense, engineering may claim one of the most brilliant achievements of this war, in the battery of 14-inch, high-powered naval guns—having a range, 28 miles, greater than any other guns in France, and mounted on railway cars—with which Rear Admiral Plunkett cut the main artery of German retreat.

So, each of the four technical Bureaus of the Navy Department could tell, if it wished, a noble story of engineering triumphs in this war, although tonight we shall have time for but one.

As to that story, I desire to express my acknowledgment to the Secretary of the Navy, through whose courtesy I am enabled to address this Society tonight, and under whose administration the United States Navy has done such effective work in the greatest war of all time.

May I add also a personal word: after my graduation from Annapolis I was for nearly twenty years in the Navy, but since then I have been long absent from that gallant service. During the Spanish-American War I was on duty in the Bureau of Steam Engineering as one of the assistants to that honored veteran of the Seven Seas and the Arctic ice, the late Admiral Melville.

When, during this war, I came to serve in a similar capacity under his successor, Admiral Griffin, I was amazed at the growth of the operations of the Bureau, both as to their scope and magnitude. From relatively small beginnings in propelling machinery, its field in dynamical engineering had become almost all-embracing, and it had expanded into a huge engineering organization, expending during the last fiscal year more than three quarters of a million dollars a day, having a staff of 137 officers and about 550 employees in Washington, and occupying in the new Navy building there nearly two acres of office space.

So—as an observer simply, and not as one who tells his own accomplishment—I may speak with full freedom of the work

of my associates of years gone by with whom, for a time, it is my honor and my warm pleasure to serve.

## THE STUPENDOUS TASK OF THE BUREAU OF STEAM ENGINEERING DURING THIS WAR.

In considering the work of the Bureau of Steam Engineering, let us glance first at the stupendous task which has confronted that Bureau during this war.

When the war began we had a Navy of about 350 ships—commissioned, in reserve, or building. But the fleet grew with amazing rapidity, and, at the close of the fiscal year on June 30 last, the Navy had in service, or soon to be commissioned, a total of 1,959 vessels. This total comprised 570 ships of the regular Navy; 93 drawn from other Government services; 937 converted merchant ships, used as troop transports, naval auxiliaries, and in patrol; and, finally, 359 vessels built by the Emergency Fleet Corporation for the Naval Overseas Transportation Service.

The aggregate horsepower of these vessels is about 6,500,000, which is thirteen times that of our fleet during the Spanish-American War. And, further, it is more than ten times the power developed on both sides of Niagara Falls, and is also equal to about one-sixth of the primary fuel and water power now employed in land service in the United States, not including the locomotives.

But the maintenance of the machinery for this huge power was not all; for at the close of the fiscal year there were under construction 376 combatant and auxiliary vessels and 52 tugs for the Navy. Included in this number is a huge force of Destroyers. When these vessels shall be completed, the present horsepower of the Navy will be increased by 70 per cent, and become 11,000,000.

Neglecting for the time the work involved in designing and building, or inspecting the building of, the engines of ships under construction, let us consider only the duties of the

Bureau with regard to the maintenance of the machinery and apparatus of the nearly 2,000 vessels in service when the war closed.

First, let me note that the Bureau of Steam Engineering is not merely a propelling-engine Bureau, charged only with keeping ships in motion. On the contrary, it is responsible for nearly all of the accessories connected with getting them under way and maintaining them in efficient condition; and, as well, for many of those appurtenances which add to the comfort and health of the crew.

For example, the Bureau must provide an abundant supply of fresh water for, in a battleship, about 1,500 men. It must also furnish machinery and appliances for ice-making and for refrigerating rooms; for electric light throughout the ship and for searchlights; for electric power used in a multitude of ways, from operating a turret to turning an ice-cream freezer; for steam and electric power for cooking, and for steam and electric heating.

The Bureau furnishes also portable electric fans for ventilating spaces not sufficiently served by the main system of the ship; also all kinds of electrical communications, including a complete telephone service; an abundant supply of water for sanitary uses; and the machinery, whether steam or gasoline, for the ship's boats.

For repair work the Bureau provides a machine shop on each vessel with facilities suitable for her ordinary needs. For heavier work it equips the shops of the repair ships which accompany a fleet. It designs and supplies the radio outfit of all ships and shore stations, including those for trans-oceanic service. For submarines it furnishes oil engines for surface cruising, and storage batteries and electric motors for propulsion when submerged. It is responsible for the machinery and wireless equipment of all aircraft, for the generation and supply of gas for observation balloons and dirigibles, and for

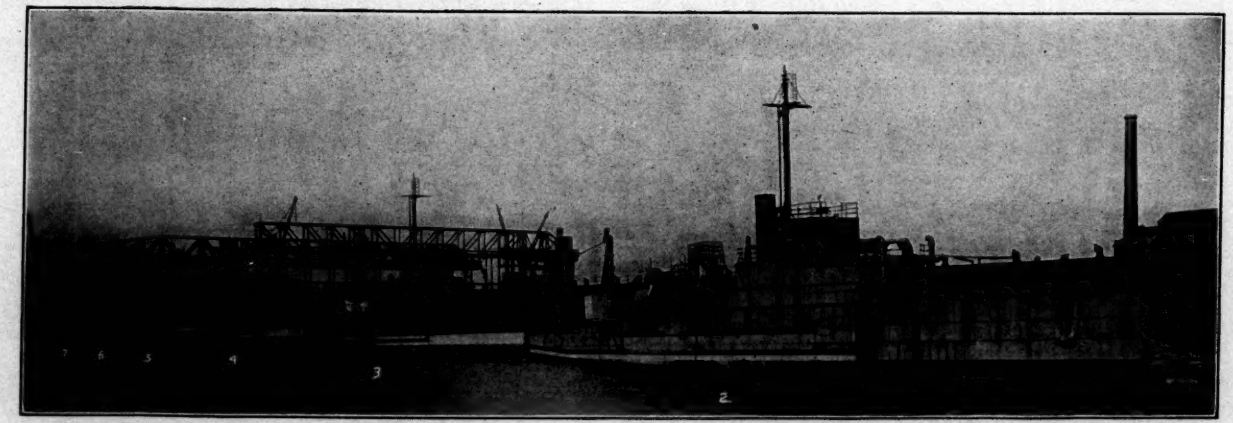


FIG. 1.—THE "EAGLES"—FORD BOATS.

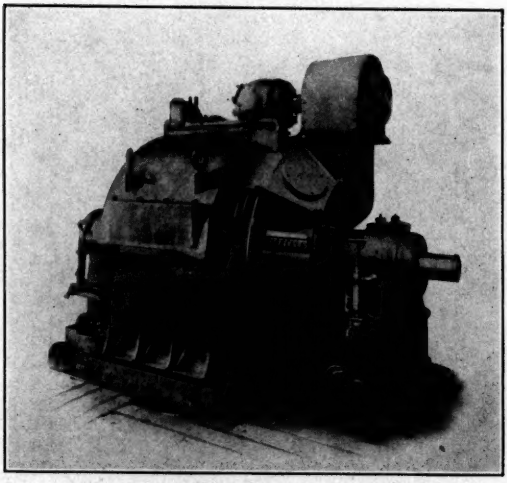


FIG. 2.—MOTORS FOR SUBMARINES.

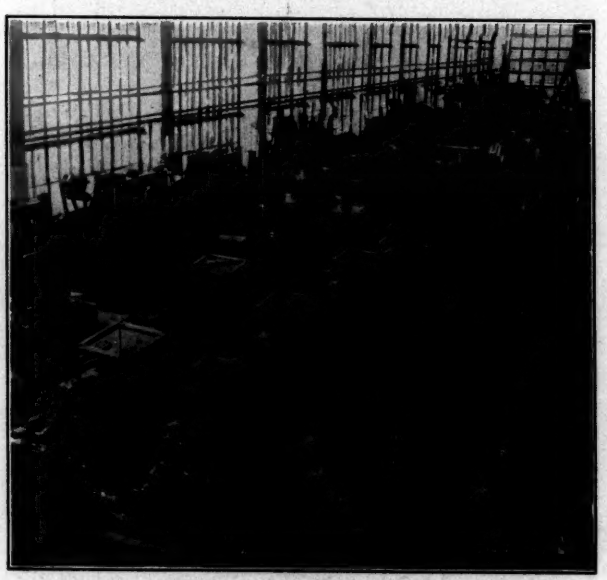


FIG. 3.—MACHINERY FOR THE "EAGLES."

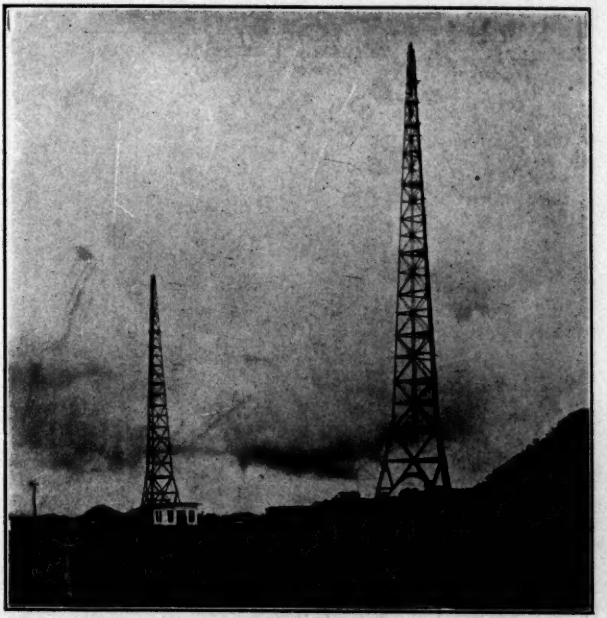


FIG. 4.—U. S. NAVAL RADIO STATION, BALBOA, CANAL ZONE.

FIG. 5.—AIRPLANE WIRELESS TELEPHONE SET.

FIG. 6.—STEAM TRAP MOUNTED ON ROCKING PLATFORM TO DETERMINE ITS BEHAVIOR ON A VESSEL IN A SEAWAY.

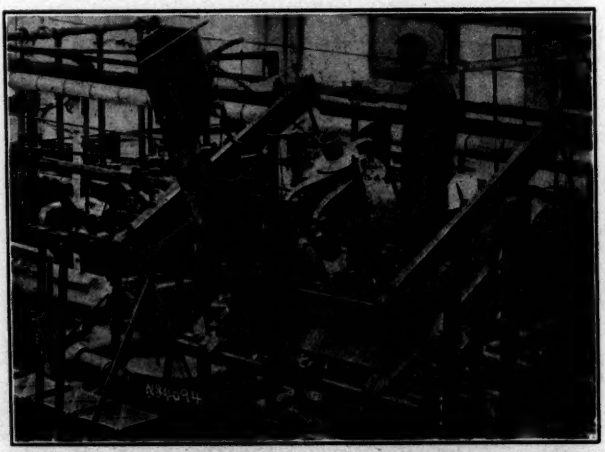


FIG. 7.—CENTRIFUGAL OIL PURIFIER MOUNTED ON ROCKING PLATFORM DURING DEVELOPMENT TESTS.

FIG. 8.—MICROSCOPIC EXAMINATION OF LARGE METAL OBJECTS. MICROSCOPE AND CAMERA ATTACHMENTS ARRANGED FOR MAKING PHOTOMICROGRAPH OF BROKEN SHAFT COUPLING.

FIG. 9.—1ST I.P. CYLINDER OF S. S. "SANTOS," NAVY YARD, PHILADELPHIA, MAY 3, 1918.

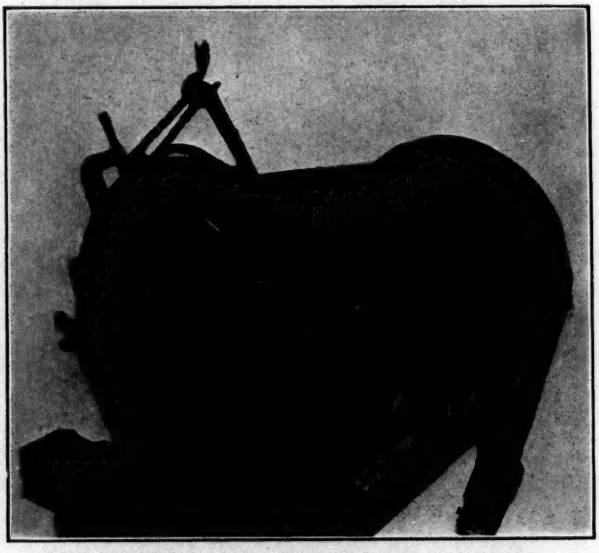


FIG. 10.—1ST I.P. CYLINDER OF S. S. "SANTOS," NAVY YARD, PHILADELPHIA; READY TO WELD, MAY 29, 1918.

FIG. 11.—1ST I.P. CYLINDER OF S. S. "SANTOS," NAVY YARD, PHILADELPHIA; FINISHED, AUGUST 15, 1918.

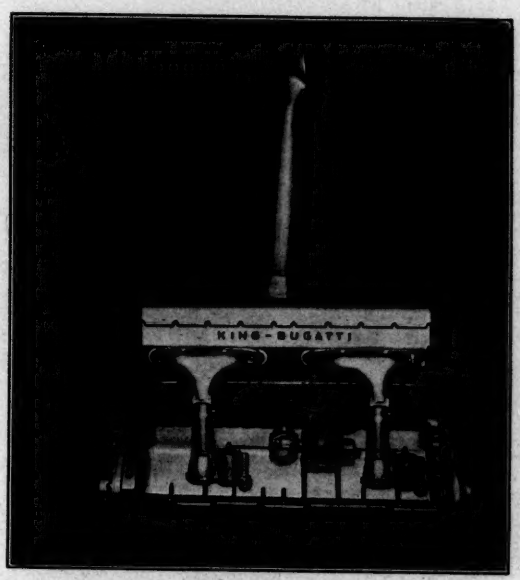



FIG. 12.—KING-BUGATTI AIRPLANE ENGINE, 16 CYLINDERS.

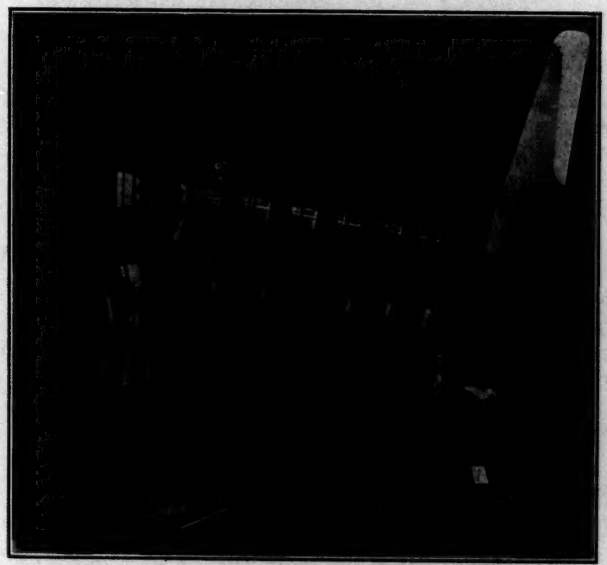


FIG. 13.—LIBERTY MOTOR, 12 CYLINDER, 400 H.P., STANDARD U. S. AERONAUTICAL ENGINE.

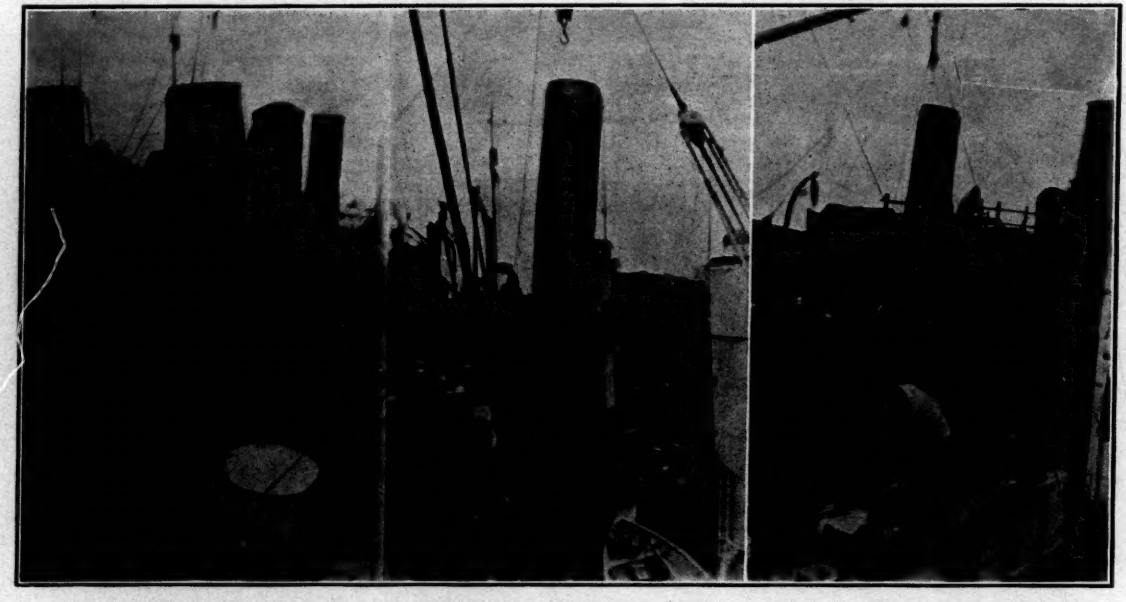


FIG. 16—STACK INSTALLED ON DESTROYER.

FIG. 15.—WHIPPING STACK TO DESTROYER.

FIG. 14.—COMPLETED STACK ON U. S. S. "DIXIE."

SMOKE PIPE MANUFACTURED ON U. S. S. "DIXIE" AND INSTALLED ON U. S. S. "AMMEN" IN TOTAL PERIOD OF SIX DAYS.

the detection devices used against enemy airplanes and submarines.

When we entered the war there was naturally a tremendous increase in the routine work of the Bureau. For the Battleship Fleet, although always ready for normal duty, needed finishing touches to fit it for possible service abroad. Also the Reserve Fleet had to be put in full commission, and the Destroyers and their accompanying Repair Ships had to be prepared and fitted out for their work in European waters.

Further, the converted merchant vessels taken over by the Government had to be repaired, altered and fitted out for use as naval auxiliaries and transports; the many merchant ships carrying armed guards had to be provided with electrical equipment—call bells, telephones, and the like—for the effective control of gun fire; and the unprecedented work of repairing the damaged machinery of the German merchant vessels we had seized had to be done at express speed. To all this there was added almost immediately the construction of a large number of submarine chasers, and later, of a huge force of destroyers and mine layers.

From this hasty outline some idea may be gained of the scope and magnitude of the work of the Bureau of Steam Engineering during the war. That, notwithstanding the enormous expansion of our Fleet in this period, it has been possible to carry on this work successfully is due to the efficiency of the prior organization of the Bureau. The Engineer-in-Chief did not create a new system, but simply expanded the one already existing, which is the development of one inaugurated thirty years ago by a former engineer officer of the Navy, Mr. Asa M. Mattice, a distinguished member of this Society.

This also may be said as to such advances as the electric drive for battleships, the exceedingly compact arrangement of machinery of 28,000 horsepower in the latest destroyers, the noiseless and vibrationless engines of the Ford boats, the progress in radio equipment and in anti-aircraft and anti-

submarine devices and other minor matters. In all these the Bureau has been successful virtually at its first attempt. And this success has not been haphazard, but was the result of careful, prior investigation and planning by an efficient organization.

In its work the Bureau of Steam Engineering has spent huge sums during this war—more than 288 million dollars, for example, during the last fiscal year. Tonight, I hope to give you, as citizens and tax-payers, a fair amount of evidence that your money has been well spent, not only by this Bureau but by all others in the Navy Department.

## ORGANIZATION OF THE BUREAU OF STEAM ENGINEERING.

For clearness in discussing the work of the Bureau of Steam Engineering let me give a brief outline of its organization and principal activities.

The Engineer-in-Chief determines all questions of Bureau policy, authorizes all expenditures, and decides all important—and many minor—questions of detail.

The Assistant to the Bureau, Captain Oscar W. Koester, U. S. Navy (retired), handles routine business in technical and administrative matters; investigates all work and expenditures in connection with the organization, building, equipment and maintenance of the shops of the Machinery Division at Navy Yards and Repair Bases, and, when necessary, inspects and reports on the condition of the Bureau's work at private plants.

The staff of the Bureau is separated into ten divisions—one clerical, headed by the Chief Clerk; and the remainder, technical. The duties and titles of the nine latter divisions are:

Design, Electrical, Repairs, Radio Telegraphy, Inspection, Supply, Fuel and Personnel, Aeronautics, and Logs and Records.

Each of these divisions is in charge of an officer who is an expert of high rank in his especial field. The muster roll of

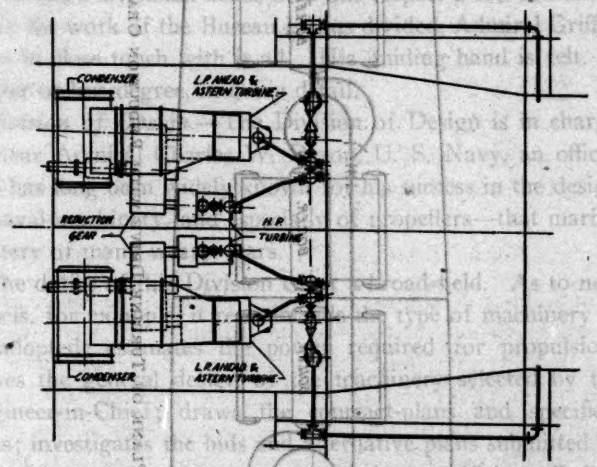



FIG. 17.—BATTLESHIP MACHINERY ARRANGEMENT; DIAGRAMMATIC PLAN OF TURBINE DRIVE WITH REDUCTION GEAR.

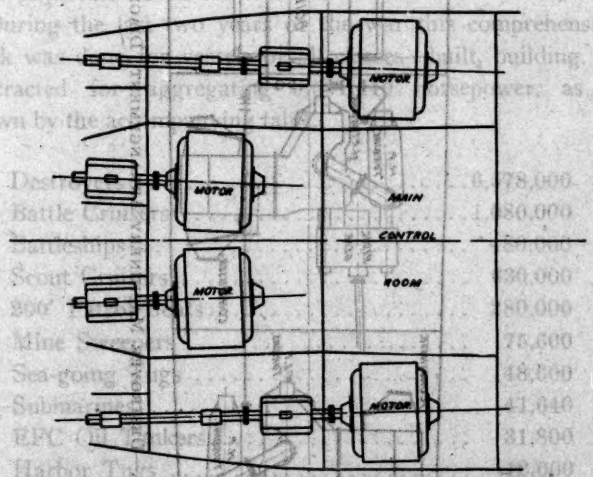



FIG. 18.—BATTLESHIP MACHINERY ARRANGEMENT; DIAGRAMMATIC PLAN OF ELECTRIC DRIVE SHOWING ARRANGEMENT OF MOTORS.

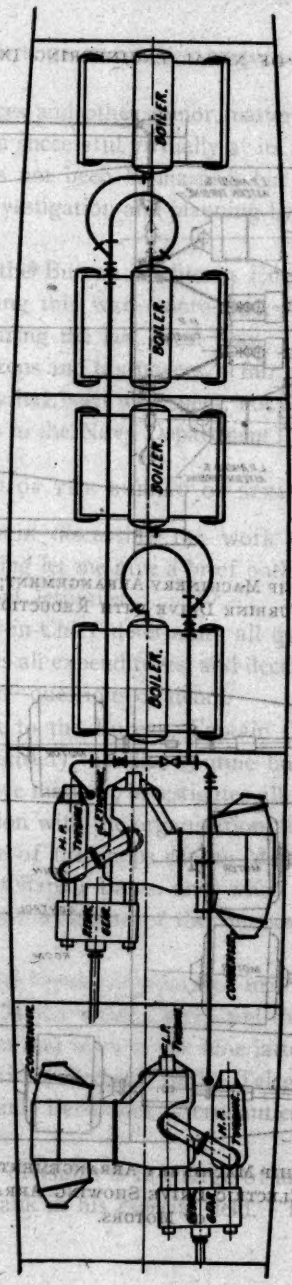



FIG. 19.—DESTROYER MACHINERY ARRANGEMENT; DIAGRAMMATIC PLANE OF TURBINE DRIVE WITH REDUCTION GEARS.

the Bureau's divisional heads is in this respect a roll of honor. While the work of the Bureau is thus divided, Admiral Griffin keeps in close touch with it all. His guiding hand is felt, in greater or less degree, in every detail.

*Division of Design.*—The Division of Design is in charge of Rear Admiral Charles W. Dyson, U. S. Navy, an officer who has long been widely known for his success in the design of naval machinery, and especially of propellers—that marine mystery of many weary years.

The duties of this Division cover a broad field. As to new vessels, for example, it recommends the type of machinery to be adopted; estimates the power required for propulsion; makes the general design of the machinery selected by the Engineer-in-Chief; draws the contract-plans and specifications; investigates the bids and alternative plans submitted by contractors, and makes recommendations; criticises the detailed plans forwarded by contractors; prepares directions for the trial of the machinery; and finally makes recommendations as to payments on the contracts.

During the last two years of the war this comprehensive work was done for vessels of all classes—built, building, or contracted for—aggregating 9,501,440 horsepower, as is shown by the accompanying table.

| | |
|---|---|
| Destroyers | 6,578,000 |
| Battle Cruisers | 1,080,000 |
| Battleships | 480,000 |
| Scout Cruisers | 630,000 |
| 200' Patrol Boats | 280,000 |
| Mine Sweepers | 75,600 |
| Sea-going Tugs | 48,600 |
| Submarines | 41,640 |
| EFC Oil Tankers | 31,800 |
| Harbor Tugs | 12,000 |
| Fuel Ships | 10,400 |

| | |
|---|---|
| Motor Tugs | 2,800 |
| Ammunition Ship | 10,600 |
| Oil and Water Barges | 2,400 |
| Gunboat | 1,600 |
| Total H. P. | 9,285,440 |
| 110′ Patrol Boats | 216,000 |
| Grand Total | 9,501,440 |

In addition to the work on new vessels the Division passes on all alterations to the existing machinery of naval vessels and those of the Naval Overseas Transportation Service. It also specifies the materials and their characteristics for marine engineering construction, and has general supervision of all tests and experiments relating to marine propulsion at the Engineering Experiment Station at Annapolis and the Fuel-Oil Testing Plant at the Philadelphia Navy Yard. Similarly, it has general supervision of the Inspection Force at building yards, a work in which the Bureau has at this time about 60 officers engaged.

*Electrical Division.*—When we entered the war the Electrical Division was in charge of Commander Guy W. S. Castle, U. S. Navy, who has been succeeded recently by Captain C. E. Courtney, U. S. Navy.

This Division supplies all the electric power, the wiring for all purposes, and, roughly, about 75 per cent of all electrical machinery and apparatus on naval vessels. Its field is therefore a most important one, extending to communication systems for the control of gun fire, the electrical signals for day and night use which are so necessary for the handling of ships in company at sea and in making recognition signals, searchlights for protection against Destroyer attack at night, the motors and storage batteries for the propulsion of submarines when submerged, anti-submarine and anti-aircraft devices, all interior communication systems, the multitude of other appli-

ances in which electricity serves on shipboard, and the electrical equipment of the Machinery Division at Navy Yards. In this work the Bureau has expended about $20,000,000 during the last fiscal year.

When the United States entered the war the Electrical Division was confronted by a huge task, since Congress had failed to appropriate sufficient funds for an adequate amount of spare material and there was little in stock at Navy Yards. For example, the amount of wire in store was almost negligible, while day by day the number of ships acquired in various ways grew with amazing rapidity.

But, six months before we became a belligerent, Commander Castle, by direction of the Engineer-in-Chief, had made a comprehensive survey of all probable requirements in the event of war, and hence he was ready to act promptly and effectively. Manufacturers everywhere were patriotically eager to aid, and so the situation was saved.

As an instance of this: Before the war wire was supplied by only two or three firms. When enormous quantities were thus needed immediately, fifteen of the largest wire manufacturers were summoned to Washington, and contracts were made with them which virtually gave the Bureau control of the wire output of this country.

In original work Commander Castle supervised the development of certain anti-submarine and anti-aircraft devices which are of a confidential nature. These submarine detectors have been made in quantity, not only for our own vessels, but for the British Admiralty and the French Ministry of Marine. For the anti-aircraft investigation Dr. G. W. Stewart, of the University of Iowa, made for the Bureau a comprehensive study at Pensacola, Florida, of all the noises made by the engine and wings of an airplane in flight. From his results and in coöperation with the Signal Corps of the Army, a scheme of sound-ranging on enemy airplanes was devised, and the necessary instruments were designed and made.

Unquestionably the most important electrical accomplishments of the Bureau during the war have been the remodeling and improvement of the wiring for gun-fire control systems in all ships from dreadnaughts to gunboats, the installation of communication systems for the service of the guns on transports and ships carrying armed guards, and the provision of recognition signals on all vessels acquired or operated by the Navy.

Not the least part of the electrical work of the Bureau is concerned with storage batteries for submarines. These serve not only for propulsion, but for nearly all auxiliary purposes, such as pumping, lighting, heating and cooling. The continuance of the war brought great growth in the number of our submarines, with consequent increases in their operations and the spare equipment that it was necessary to carry. All of our submarine batteries are of the lead-acid type. In some cases the expedient was adopted of reducing the thickness of the plates in order to increase the exposed surface. While the ultimate effect of this is to reduce the life of the battery, the military advantage gained in thus obtaining one of greater power for the same weight, was considered sufficient to offset the disadvantage of short life.

Owing to the magnitude of the work of the Electrical Division and the fact that so many regular officers of the Navy who were qualified for this work were ordered to sea duty, the Bureau succeeded in having one hundred graduates in electrical engineering enrolled in the Naval Reserve and given a short course in training for naval work. These young officers have been serving very creditably in the Fleet.

*Supply Division.*—The Supply Division is in charge of Lieutenant Commander Charles K. Mallory, U. S. Navy (retired), a member of this Society, who, on the mobilization of the Navy for war service, was assigned to this duty as the relief of an officer on the active list.

Matters relating to engineering materials and supplies—

under cognizance of the Bureau of Steam Engineering—for vessels of the Navy and for Navy Yards, are handled through this Division. The specifications for the material are drawn by the technical Division concerned, and all subsequent questions of a technical nature are referred to that Division for decision. Hence, the Supply Division is, in some respects, a sort of center of business activity for the Bureau in its contact with bidders and contractors. During our first year in the war it passed upon materials aggregating $100,000,000 in value, and this was for miscellaneous materials only, and did not include new construction work. The actual purchase is made by another Bureau, which refers to the Bureau of Steam Engineering all bids for determination as to the one that is most advantageous.

Under peace conditions—when contracts for the construction of vessels as well as the purchase of supplies were made on a fixed price basis, with a penalty for failure to complete or deliver on time—the duties of the Division consisted primarily of passing upon requisitions for materials as to their quantity, quality and suitability for the purpose intended, and of analyses of the proposals received from bidders with recommendations as to the award of contracts. In addition to this service the Division was charged with the duty of providing suitable allowance lists, covering spare parts and material for the outfitting and operation of vessels of the Fleet.

Upon our entry into the war it soon became apparent that the question of obtaining material and supplies—both for the direct use of the Navy and for those having contracts for new construction—would be of increasing importance and difficulty. Owing to the unsettled conditions as to materials, supply and labor, it became necessary to abandon the penalty for failure to complete contracts on time, and in many cases contracts had to be changed from a fixed price basis to one which would insure the receipt by the contractors of a reasonable profit.

Many new Government activities, as well as others of a private nature closely associated therewith, came into existence, requiring material in excess of the country's output. The War Industries Board and other Commissions and Committees, with which we are all more or less familiar, were formed to control the resources of the country and to direct material into channels where it was most needed. All of this meant that the Navy's engineering requirements had to be closely studied in order to insure delivery of material when and where required, and to avoid the making of unnecessary demands at the expense of other essential war activities.

Owing to the change in the form of contracts, various questions were continually arising as to which the Bureau was called upon for an opinion or a recommendation. The handling of such situations was centered in Washington, the Supply Division being expanded as occasion required to take care of the increasing volume and variety of work. A follow-up system for all requests for material was also instituted, by means of which the exact status was shown at any time up to the actual completion of contracts.

Under peace conditions, after the placing of a contract for engineering material, either for the direct use of the Navy or for Government contractors, it was inspected as to quality before delivery by the various Naval Engineering Inspectors throughout the country. Also, it was unnecessary in those days to give much attention to the matter of the ability of the contractor to deliver as required. This latter consideration, however, became more and more important in war time, and, to take care of the situation, a number of business men of ability, who were willing to devote their time to Government needs, were enrolled as officers in the Naval Reserve Force and assigned to production work in the various Inspection Districts.

The duty of these officers was to follow closely every contract; to know whether it would be filled as required; and, if

not, why—so that any necessary action could be taken promptly. In addition to the regular Inspecting Force of the district, who aided in this work, enlisted men were also provided by the various Naval Districts. The work of these production officers was of inestimable value in obtaining material as required.

The Supply Division has also made a revision of the engineering allowance lists for all regular Navy vessels, to insure their being on a proper war footing in this respect. It will be seen that, owing to the variety and volume of its work, this Division has had its hands full since our entry into the war.

*Radio Division.*—Commander S. C. Hooper, U. S. Navy, is in charge of the Division of Radio Telegraphy. The activities of the Bureau of Steam Engineering in this respect cover almost a world-wide field. All matters relating to radio equipment—except the actual operation by the radio personnel—on vessels operated by the Navy, on all vessels of the Army and other branches of the Government, including the United States Shipping Board, and on merchant ships requisitioned by that Board, are directed by the Bureau. Virtually, therefore, the radio installations on all vessels flying the American flag is, in these respects, under the direction of the Bureau of Steam Engineering. The number of these installations now exceeds 4,000.

The Bureau's direction extends similarly to 50 naval radio coastal stations and 75 commercial coastal stations in this country; to others in the West Indies and the Canal Zone, in our island possessions and Alaska and Vladivostok in the Pacific; and, finally, to one now building for us at Bordeaux in France, which will be the most powerful wireless station in the world.

Our Atlantic, Pacific and Gulf Coasts, and the shores of the Great Lakes, are covered by strategically located low-power and semi-high power radio stations. Washington is in touch also with our possessions in the West Indies by means of the

It should be noted that the weight measurement is not a trustworthy gage of the amount of work done in inspecting material. For example, tons of ingots, pig iron, or forging billets may be examined and accepted by an inspector in a single day, while several days may be necessary to inspect certain aircraft material or delicate machinery weighing but a few pounds.

*Fuel and Personnel.*—The Fuel and Personnel Division is in charge of Commander H. A. Stuart, U. S. Navy. During the greater part of the war, it was headed by Commander N. H. Wright, U. S. Navy.

Fuel oil, gasoline and lubricating oils are vital essentials of modern navies, hence the work of this Division is of fundamental importance, especially as the Navy is now hard pressed in its effort to retain the naval oil reserves in California. However, all of the Navy's new ships are oil burners, since, during their lifetime, fuel oil for them will not fail.

This Division covers a broad field. During the last fiscal year it assisted in extending the coaling facilities in our ports and in making a comprehensive survey of the low-volatile steaming coals with the view of locating mines capable of furnishing suitable fuel for the Overseas Transportation Service. It modified the naval specifications to permit the use, for a time, of a considerable percentage of Mexican distillate, and made preliminary investigations for establishing specifications for heavy fuel oil for transports and cargo carriers. Further, it coöperated with various Government agencies in the adoption of standard specifications for aviation gasoline.

The Division has also coöperated with the Bureau of Navigation in the effort to obtain officers qualified for engineering duty in the regular Navy and in the Naval Reserve. Virtually all of the nearly 2,000 ships built and to be built by the Emergency Fleet Corporation are to be operated by the Navy, a service which will require a total of about 25,000 deck and engineer officers—and that is another matter in which the Navy is hard pressed.

*Division of Repairs.*—The Division of Repairs is in charge of Commander W. A. Smead, U. S. Navy, who has been for some months in Europe. In his absence it is headed by Lieutenant Commander Bruce.

Next to importance to building a ship is keeping it in repair. Lame ducks cannot fight. During the war this Division has had its hands full with the maintenance of, and the repair work for, the machinery of the Fleet and the vessels operated by the Navy—a total of nearly 2,000 by the end of last June. However, it measured up fully to all requirements. No vessel of importance has been laid up for repairs. In addition to this it has handled all matters connected with the taking over and fitting out of vessels acquired or to be operated by the Navy.

Since we had more than 300 vessels in European waters, the problem of their repair and maintenance was not simple. Spare parts had to be ready and matériel needs anticipated. To this end a stock of the most important engineering material was provided by this Division—some in this country, but the greater portion at the foreign bases—and its location and use so thoroughly systematized that it could be shipped upon cable advices.

It is noteworthy in this connection that when one of the Destroyers was so badly damaged by a submarine that it was with difficulty that she was able to make port, spare shafting and propellers were ready for her long before their installation was necessary. In another case, where it was seen that the boilers of a Destroyer needed retubing and the time required would have put her out of service for too long a time, new boilers were built and, when ready, were installed with a surprisingly short period of inactivity.

For this work of repair and supply the Navy established a total of five major bases—three in France, one in Great Britain, and one at Gibraltar. There are also two minor bases in or near the Mediterranean. The facilities of these bases were reënforced by those of six Repair Ships, general plans of

whose shops are shown in the views. These vessels proved to be of the greatest value to a fleet in a war area, and their work was frequently commented upon, and very favorably, by both Vice Admiral Sims and the officers of foreign services. Without these ships, our Destroyers could not have maintained the wonderful record they established of being always ready. In this respect our Navy leads all others.

Our Navy had an average of approximately 50 Destroyers serving for 19 months in European waters. One Repair Ship there re-tubed two Destroyer boilers (Normand) in nine days, the boilers being in place. The services of these Repair Ships were so effective that only $1,100,000 were spent on this whole Destroyer Force for machinery repairs made by other agencies during this period, and this total covers several cases in which there were very serious damages by torpedoes or collisions.

As illustrating the resource of the Repair Ships, the retubing of boilers of a Destroyer was undertaken without interruption of her regular patrol and convoy duty by working on one side of the boiler during her 5-day overhaul period and completing the other side during the following period, and so on until all were finished, the vessel meanwhile being capable of 25 to 28 knots speed.

It is doubtful whether such a performance would have been possible under other direction than that of Vice Admiral Sims, who insisted that Destroyers must have a regular overhaul period uninterrupted by any other duty or consideration.

*Division of Aeronautics.*—The Division of Aeronautics is in the charge of Commander A. K. Atkins, U. S. Navy. The work of this Division covers aircraft machinery generally, engineering design for this purpose, and the generation and supply of gas for observation balloons and dirigibles.

The marked military advantage to be gained by having but one type of airplane motor prompted the Navy Department to adopt the motor used by the Army. All Liberty motors for the Navy are, therefore, obtained through the Aircraft Pro-

duction Board. Orders have been placed for more than 4,000 of these Liberty engines, delivery to be completed by January 1, 1919. Of this number about 1,300 were received by the end of the last fiscal year, and distributed to various naval air stations in this country and abroad. There have also been purchased about 500 Hispano-Suiza engines for use at naval air stations in this country.

At various times the Bureau of Steam Engineering has tested and used a total of eleven airplane engines as follows:

| *Motor.* | *H.P.* | *Weight, Lbs.* | *Remarks.* |
|---|---|---|---|
| Liberty, 12 cyl..... | 380 | 853 | Low comp. water cooled. |
| Geared Liberty, 12 cyl..... | 450 | 972 | V-45°; water cooled. |
| Kirkham, 12 cyl..... | 400 | 670 | V-60°; water cooled. |
| Dusenberg, 16 cyl..... | 800 | 1,617 | V-60°; water cooled. |
| Bugatti, 16 cyl..... | 475 | 1,170 | Cyls. in parallel rows of eight; crank shafts geared together. |
| Kessler supercharger..... | 600 | 600 | 6-cyl., vertical. |
| Hall-Scott, Liberty Six..... | 213 | 490 | Vertical, water cooled. |
| Union Engineer, 6 cyl..... | 125 | 443 | Vertical. |
| Hispano-Suiza, 8 cyl..... | 150 | 440 | V-90°; water cooled. |
| Curtiss, OXX-6, 8 cyl..... | 100 | 395 | V-90°; water cooled. |
| Curtiss, V-2, 8 cyl..... | 300 | 690 | V-90°; water cooled. |

Among the more important of these motors are: the Liberty, 12 cylinder; the Geared Liberty, 12 cylinder; the Hispano-Suiza; the Union Engine; and the Kirkham Motor.

In general engineering design the Bureau has done much work, especially with regard to the application of the Liberty Motor to seaplanes. This work covered starters and priming devices; the development of a flow meter, pneumatic-spark and throttle controls, and an oil-cooling system.

One of the very important duties devolving on the Bureau of Steam Engineering is the equipment and maintenance of stations for the generation and supply of a non-inflammable gas for use in observation balloons and dirigibles. Somewhere in the United States there is a plant for this purpose, on which the Bureau was authorized to expend $3,000,000. A number

of stations have been established and a full equipment of gas-containers provided, so that any calls may be promptly met. This gas has been shipped in quantity to the British and French. For this balloon-gas service it has been necessary to give suitable training to a number of young officers and to assign them to stations at home and abroad.

*Division of Logs and Records.*—The Division of Logs and Records is in charge of Commander W. W. White, U. S. Navy (retired). In the effect of its work on marine engineering design and progress, the field of this Division is of great importance.

Its main functions are: the checking up and review of the engine-room logs of naval vessels; the compilation and issuance to the seagoing personnel and naval training schools of Bureau pamphlets relating to the care and management of machinery on shipboard; the preparation and issuance to the naval service of the "Confidential Bulletins of Engineering Information;" and the filing and indexing of all important data regarding engineering matters, with especial reference to naval vessels.

It will be seen that the work of Commander White—an officer of long experience in this duty—is of marked importance, not only to the Navy, but to the engineering profession.

## NAVY YARDS AND SHORE STATIONS.

The efficiency of the Machinery Divisions at Navy Yards and Shore Stations has a most important effect on the general success of the Bureau's work, in keeping ready for service the machinery of the Fleet and of other vessels operated by the Navy, and also in building machinery for new vessels.

During the last fiscal year, in nine Navy Yards and Naval Stations in the United States and our insular possessions, there were employed on Bureau work exclusively about 175 commissioned and warrant officers and 9,700 mechanics. In four other Navy Yards, under Industrial Managers who direct the

work for all *matériel* Bureaus, about 110 commissioned and warrant officers and 11,500 mechanics were engaged in these activities.

An adequate resumé of the accomplishment of these large forces during this war would take far more time than we have available this evening. However, as an example, I may give a few particulars as to the immense amount of work carried on at the New York Navy Yard under the direction of Rear Admiral George E. Burd, U. S. Navy, Industrial Manager. Since April 6, 1917, Admiral Burd and the officers serving with him have had charge of the conversion and repair, in all departments—including the mounting of batteries in many cases—of a total of 723 vessels of all classes. Much of this work was under cognizance of the Bureau of Steam Engineering.

In addition to this, the propelling machinery, in whole or in part, of three battleships has been building there, and considerable other new construction work of a minor character. Further, there has been a great deal of electrical and radio work for vessels of all classes. The number of men employed—excluding the clerical, drafting, and sub-inspector force—was about 4,900.

In connection with the Navy Yards, a very large amount of work has been done by private establishments on naval vessels and transports. And in the prosecution of this work the closest coöperation prevailed between the Commandants of the various Naval Districts and the Commandants of Navy Yards and the Bureau of Steam Engineering, and also with Vice Admiral Gleaves, in command of the troop transport service. Without the team work that thus prevailed and the harmony of interest ever uppermost, many a sailing would have been delayed.

### NAVAL EXPERIMENT STATION AT ANNAPOLIS.

The Naval Experiment Station at Annapolis does much of the marine engineering research and experimental work of the

Bureau. The Head of this Station is Rear Admiral T. W. Kinkaid, U. S. Navy.

The services of this Station are of the greatest value, not only in the designing work of the Bureau, but in the maintenance of the machinery of the Fleet. Its chief function is the prosecution of research work on, and making tests of, appliances, materials, and methods which relate to the propelling plants of naval vessels, such as forced-draft blowers, pumping machinery, feed-water heaters, coolers, refrigerating machinery, steam turbines, condensers, evaporators, boilers, coal and oil fuels, packings, and so on. The Station's activities also include much work in the metallography of steel and other metals.

### FUEL-OIL TESTING PLANT.

The Fuel-Oil Testing Plant at the Philadelphia Navy Yard has also done a very considerable amount of valuable work during the war. Its present head is Lieutenant A. M. Penn, U. S. Navy. Its field covers burners, air registers, the refractory linings for furnaces, furnace insulation, and other similar matters.

During the last fiscal year, in order to obtain comparative data of boilers of various types, a test of the White-Forster boiler was conducted at this plant. The results show that express type boilers of this kind, having a large amount of heating surface and a thick tube bank, are even more efficient, at from low to moderately high rates of combustion, than the straight-tube boilers now used so extensively in our Navy, but that they are slightly less efficient at very high rates. The express type has advantages also in the saving of weight, original cost, and cost of upkeep.

### SOME RECENT DESIGNS.

Let me cite a few noteworthy instances of recent designs, and of research and development work, by officers of the Bureau of Steam Engineering.

*Propellers.*—The diagram, showing the estimated and actual performance of a battleship propeller, is largely typical in indicating the recent progress made by Rear Admiral Dyson in solving this complex problem.

*Destroyers.*—Immediately after the declaration of war by this country, our building facilities were concentrated on light craft—destroyers, submarines, patrol boats, and mine sweepers—to meet the menace of German submarines and mine layers. The figure shows diagrammatically the arrangement of turbines with reduction gears for the propelling machinery of the destroyers. This arrangement is unusual in the compact installation of engines of 27,500 horsepower on an average floor space of 26 x 56 feet.

*Patrol Boats, "Eagles."*—Because of their chief duty—hunting submarines—the machinery of the "Eagles," built by Henry Ford, was required to be as smooth running, vibrationless, and noiseless as possible, in order to secure the most favorable conditions for "listening" in submarine detection. This object was attained in the Bureau's design by omitting reciprocating parts almost wholly in the main and auxiliary machinery, nearly all motion being rotational. An officer who was present during the trial of the first of these boats, says that it gave him a peculiar sensation to be on a vessel speeding at 18 knots with so little sound and vibration.

## RESEARCH AND DEVELOPMENT.

Some very striking advances of high military value in radio and submarine apparatus and in aircraft equipment are due directly to the research and development work conducted under direction of the Bureau of Steam Engineering during this war.

*Submarine Detectors.*—Since the war began eight devices for submarine detection have been employed by the Bureau and made in quantity, not only for our own Navy, but for the British Admiralty and the French Ministry of Marine.

Modified forms of these devices have also been designed for use in the protection of the Atlantic coast of the United States.

In the experimental investigation of this problem the scientists who worked on it under the direction of the Bureau of Steam Engineering realized from the beginning that the noises emitted by the submarine gave the most feasible means of locating it, since water is an excellent conductor of sound waves. Therefore, virtually all of the instruments thus developed and used to hunt German submarines are acoustical in nature. The successful results which have been obtained show the wisdom of research and invention along these lines.

The chief desideratum was, of course, to produce a device which would enable the pursuing vessel to determine the location of the submarine and then destroy it. It turned out, however, that the chief function of these devices was to render the submarine powerless, rather than to make its destruction possible. When the commanders of German submarines learned, through bitter experience, that, as long as their motors were running, the location of their boats could be ascertained by any nearby vessel equipped with detection apparatus, they ceased taking hazardous chances.

Instead of going where they pleased and making surprise attacks at will, they found that it meant almost certain destruction to keep underway in the vicinity of enemy vessels. The result was that German submarines spent a large share of their time resting on the bottom, where they were as incapable of doing harm as if they had never left their docks. It is apparent, therefore, that the number of submarines actually destroyed is not an accurate index of the efficacy of these submarine detection devices.

Briefly, the reason why a submarine must lie motionless to avoid detection when the enemy is near, is that the underwater boat when underway gives out a distinctive noise which can be recognized at once by the listener stationed at the detection device of the hunting vessel. This noise is similar to that

made by a phonograph when the record has been played and the needle is traveling through the inner grooves of the disc. It has been often noted also that a submarine emits a periodic "squeak." These and other distinguishing characteristics make it possible for the listener to detect a submarine, even in the presence of other vessels underway at the same time.

Obviously, since the matter is a military secret, the various forms of anti-submarine devices cannot be explained in detail. It may be said, however, that the first types of these apparatus were rigged outboard, and that, in the second stage of their evolution, the devices were towed astern or rigged through the hull. All of these methods, however, proved inadequate, since it was impossible for the hunting vessel to use the apparatus while underway. More recent experimentation has developed a device in which the "ears" are made an integral part of the hull of the vessel, and hence will serve while underway.

Considerable progress in the perfection of these devices was being made when hostilities ended. While the ideal detection instrument has not yet been evolved, it is wholly probable that further experiments, now under way, will result in the development of devices as far in advance of the present types as the latter exceed the first crude detection instruments.

*Helium.*—Helium is the mysterious gas which, camouflaged as "Argon," the Bureau of Steam Engineering and the Bureau of Mines of the Interior Department strained every nerve to produce in quantity during the last six months of the war.

The hydrogen employed for floating observation and dirigible balloons has one vital defect—its inflammability. When struck by an incendiary bullet the balloon instantly bursts into volcanic flame and falls in smoking fragments to the earth. Helium, on the contrary, is an inert gas like nitrogen. Its use in balloons means, therefore, that the fire risk is eliminated, and that a helium-filled airship could be brought down or seriously injured only by driving an airplane into it.

Helium is a relatively rare gas, and it is both difficult and

costly to separate it from the mixtures in which it occurs in natural deposits. Until the year 1918 there had never been more than five cubic feet of helium in any one container, and its cost had been from $1,500 to $6,000 per cubic foot. Our Government is now producing it at a rate of about $80 per thousand cubic feet, and shipments for use in kite balloons for the Army had been made just prior to the armistice.

Investigations, made prior to the war, showed that considerable quantities of helium existed in the natural gases of Kansas, Oklahoma and Texas. In the Province of Ontario, Canada, an appreciable amount of helium was also found in certain deposits of natural gas there. In any case, the most practicable means of separating the helium from the other components of the natural gas was the liquefaction of all of the gases except the helium, which could then be drawn off. This process involved the use of apparatus for obtaining temperatures as low as 317 degrees F. below zero.

While the United States remained neutral nothing was done with our deposits of natural gas containing helium. But as soon as we entered the war the British Government referred to our military and naval authorities various research problems whose solution would aid in ending the war. Among those problems was that of securing an adequate supply of helium for use in observation and dirigible balloons.

This question was considered by the Joint Army and Navy Aircraft Board, and the result of the deliberation was that the Bureau of Mines began the design of a plant, on original lines, in the endeavor to lessen the cost of separating helium, while the Bureau of Steam Engineering of the Navy Department insisted on modifying existing apparatus and, so far as possible, on securing the help of operating companies in obtaining the gas.

So the effort to produce "Argon," the camouflaged helium, was prosecuted vigorously. The Bureau of Steam Engineering enlisted two organizations in the matter, the only ones in

this country which had the necessary experience for such novel work. The Aircraft Board had made recommendations in October, 1917, and, in March, 1918, the first of these organizations had a plant in operation. This was followed in May by the completion of a unit by the second company. During the experimental period of the operation of these plants the Navy Department has been in close touch with all that was done and with all organizations which could aid in the work.

Experimental work has continued steadily with the earlier plants and that of the Bureau of Mines, which was completed in October, 1918. The production of helium in quantity was recommended by the Aircraft Board in August, 1918, and, when this recommendation was considered by the Secretaries of War and the Navy, it was decided that one agency should do the entire work, the Navy Department being the one designated. Since that decision was reached a production plant has been designed, under direction of the Bureau of Steam Engineering, by the organization which built the first of the two experimental plants noted above, and the construction of the buildings at Fort Worth, Texas, has been undertaken by the Bureau of Yards and Docks of the Navy Department. A pipe line to supply the plant with natural gas is also being laid. The first unit of this quantity-production plant should be in operation by next March. So America has solved the problem of helium for aircraft and observation balloons.

*Submarine Radio Equipment.*—A number of marked advances in Radio Telegraphy are due to the action of the Bureau of Steam Engineering. The work on the various systems described below has been done largely by outside parties in coöperation with the Bureau.

For example, take the radio equipment of submarines. The underwater radio equipment for these boats consists of a single-turn, insulated loop antenna. The loop is made up of two highly insulated wires, one of which is grounded to the

hull at the bow and leads aft to the conning tower and then down through a deck insulator to the radio room. The other wire is grounded to the stern of the submarine and leads forward to the conning tower and thence to the radio room. This constitutes a closed loop having the hull for a return wire. This loop is connected to the standard radio equipment in the usual manner, except that a condenser is used in series with the loop when transmitting with the loop. It is possible to receive under water at depths reaching 20 feet by this apparatus. Sending under water has not so far been accomplished.

*Subterranean Radio System.*—The underground radio system experimented with, and from which the developments were made, was that of Mr. J. H. Rogers, of Hyattsville, Md., and consists of highly insulated wires laid in moist earth or water. These wires are laid at the four points of the compass from the receiving station, the outer ends of which are insulated. The ends of the wires in the receiving station lead up to a selector switch. This switch is connected to the receiving set, by the use of which various combinations of wires can be obtained. The underground wires are of different lengths for the different wave lengths. The standard receiving apparatus is connected in the usual manner. Efficient reception with this system is possible; it has directional qualities.

*Radio Compass.*—Let a wireless wave strike a rectangular loop, kept in a vertical plane and having a detector in its lower side. Also let the loop be capable of rotation on a vertical, central axis. Then, if the plane of the loop is perpendicular to that of the wave, the currents induced in it will neutralize each other, and the detector will register no sound. But, if the plane of the loop is at any other angle with that of the wave, the induced currents in its vertical sides will have the same direction, and a sound will be received through the detector.

This is the basic principle of the Radio Compass. In using it, the loop is swung, first, in one direction until the sound received from the detector is loudest; then in the other direction until the sound received there reaches a maximum; and, finally, the angle between these two maxima is bisected, and the plane of the loop if set at this mid position will be pointing directly at the station from which the wireless wave emanated.

This compass has been so perfected that it is possible to steer a ship into port in any weather by its use. Recently, one of our naval vessels got its position in foggy weather from the station at Cape May, 280 miles distant.

*Radio for Aircraft.*—Radio telegraphic apparatus for seaplanes has been developed which is efficient for ranges up to 200 miles, and for dirigibles up to 600 miles. Radio telephone sets, having ranges one-half those of the telegraphic sets, have also been developed. A trailing wire is used as an antenna.

Communication between planes in flight is possible up to ranges of 75 miles by wireless telegraph and 50 miles by telephone. The most satisfactory radio telephone for seaplanes is the product of the General Electric Company. Radio telephones have been installed on virtually all our naval vessels, the type used being principally that furnished by the Western Electric Company. The radio telephone has been especially valuable for manoeuvering hunting squadrons of submarine chasers and seaplanes.

A radio compass has also been developed by the Bureau engineers for the purpose of determining direction by radio signals received in an airplane. The instrument gives fairly accurate direction up to 500 miles when the plane is in flight and is exceedingly valuable for aircraft navigation.

## REPAIR OF THE GERMAN MERCHANT SHIPS DAMAGED BY THE VANDALISM OF THEIR CREWS.

The almost incredibly rapid repair of the German merchant vessels lying in our ports and seized by our Government when

we entered the war makes one of the most striking and dramatic stories in the history of naval engineering. I am unable to give it in full, since the Secretary of the Navy will treat the matter at some length in his forthcoming Annual Report, and I may not anticipate his statement. I will, however, give some details—engineering mainly—which will probably not be embodied in that report.

The views which will be shown comprise: one showing the use of oxyacetylene welding in the repair of some of the large main cylinders of these ships, and others illustrating the repair by electric welding of broken parts of the machinery of the ex-German steamer *Santos*.

The *Santos*, I should state, is not one of the vessels seized by our Government. When war broke out, 46 German merchant ships were interned in Brazilian harbors. When Brazil entered the war its government seized these ships, but not before some damage had been done to their machinery by their crews. Later, Brazil turned over 29 of these vessels to the French Government. One of them, the *Santos*, could not be repaired with the facilities available at Rio Janeiro, since her machinery was virtually a heap of junk.

So the French naval attaché at Washington appealed to our Navy Department, and, at the suggestion of the Engineer-in-Chief, the broken parts of the *Santos* machinery were crated and shipped to the Philadelphia Navy Yard, where they were repaired by electric welding under the supervision of Captain Clarence A. Carr, Engineer Officer of that yard, through whose courtesy I am enabled to show the *Santos* views tonight. Their value lies in the fact that they form a complete series, illustrating fully the methods used by our Navy in repairing the ships we seized, except that, in most cases, our work was done on the broken cylinders while they were in place and the engines were not dismantled.

There were, in all, 103 of these German ships seized by our Government. Of this number about 50 were turned over to

the Navy by the Shipping Board for repairs. The major damage inflicted was the breaking of cast-iron parts of the main engines, chiefly the cylinders. But, in addition, some connecting rods, piston rods, and boiler stays were sawed through, boilers were burned out by dry firing, and there was much minor vandalism.

The chief problem, however, was the repair of the broken cylinders, some of which were more than nine feet in diameter. The Bureau estimated that with such facilities as were available for this heavy work it would take at least eighteen months to replace the damaged cylinders by new ones. Meanwhile there was bitter need of rapid transport for our troops to France. On the removal of that crisis there seemed to depend the salvation of Christendom.

At this juncture, Captain Earl P. Jessop, then Engineer Officer of the New York Navy Yard, recommended that repairs be made by welding, and in this recommendation he was heartily supported by Rear Admiral George E. Burd, Industrial Manager of that yard. The Engineer-in-Chief at once ordered Captain Oscar W. Koester, Assistant to the Bureau, to New York to investigate the situation fully. As the result of his report the Bureau immediately issued orders to make repairs, where possible, by electric or oxyacetylene welding, and where welding was impracticable owing to the location of the break, to resort to mechanical patching—that is, to "soft patches" secured by body-bound bolts.

From this stage onward the Engineer-in-Chief placed the repairs under the personal supervision of Captain Koester, who in his resourceful ability and driving power is unsurpassed. His work went on night and day, and in five months he traveled about 14,000 miles in railroad trains from one navy yard or private plant to another along our Atlantic coast. At the expiration of that period, his huge work was done and the ships were ready for service.

As soon as repairs were completed on each vessel the

strength of the new seams was tested under the full pressure of the original design, and none failed. She was then sent to sea on a minimum trial run of 48 hours under full power, the instructions to her officers being to break down the machinery by legitimate over-work, if they could. Every engine on every vessel withstood all tests, and the ships have been in continuous service since then.

Some few of these large cylinders—about ten, I believe—were repaired by oxyacetylene welding. This method was abandoned, however, not because of the quality of the weld, but because the method did not suit the conditions. With it the cylinder had to be dismantled and removed, and then, owing to the size of the flame, the huge mass of metal had to be pre-heated. These conditions did not exist with the electric arc, whose welding point is very small and can be localized, and whose temperature can be absolutely controlled.

As to the employment of the electric arc for this purpose, this method is too well known for description here. Its use, however, on such an extensive scale was unprecedented, and great credit is due the staff of the welding company who did so rapidly such perfect and, at the time, such vital work, and also to the officials of that company, who personally coöperated with the Navy in this great undertaking.

The rapidity of getting these ships ready for service was facilitated by one other action. On many of them, in addition to the damage done by vandalism, extensive repairs were necessary, owing to their deterioration from long idleness. As the necessary overhaul and all routine repairs could be made by a capable naval crew under able officers, the Engineer-in-Chief laid the matter before Admiral Benson, Chief of Operations, with the request that the ships be fully commissioned before repairs were begun. This was done, and thus while the welding operations were in progress the ships were prepared in all other respects for sea.

One other feature of this matter deserves comment, and

that is the military value of this work in the transport of troops. These fifty ships were in service for about a year before hostilities ended, and this is approximately the time saved in these repairs by using welding methods. Twenty of these vessels can carry about 70,000 troops in one trip, and ten round trips a year is a conservative estimate of their performance. Representative Young, of North Dakota, who crossed the Atlantic in the *Leviathan*, stated recently in the House that she had transported more than 99,000 troops up to August 20. So it may be justly claimed that the rapidity of these repairs had a marked effect on the early end of the fighting in France.

It is true that criticism was leveled at the Bureau for using these welding methods, and, too, by not a few engineers and shipbuilders. But the answer to all criticisms from every source is *that the ships are running*, without accident or delay.

## THE ELECTRIC DRIVE FOR BATTLESHIPS.

In those stormy days toward the end of the Civil War, Isherwood, that master engineer, designed for the United States Navy the 17-knot cruisers *Wampanoag* and her sisters, which, as their trials proved, were incomparably the fastest ships, naval or merchant, at that time in the world. The jealousy and ignorance of those days of semi-sail power brought down on him a storm of criticism, and ultimately the great Engineer-in-Chief was forced to see his peerless ships rot at their moorings.

In the years since then there has been no fit comparison with this shameful episode in our naval history, except, in less degree, the similar storm which burst on the head of the present Engineer-in-Chief, when, with high courage and far vision, he urged the introduction of the electric drive for the propelling machinery of battleships and battle cruisers.

In one vital element, however, there was a radical difference between these two historic controversies. Isherwood was

faced by a preponderating opposition from the officers of the old sailor Line of the Navy, and, on the contrary, the adverse criticism as to the electric drive had no naval origin. By far the great majority of the Navy—its officers and officials—stood to a man behind the Engineer-in-Chief in his steadfast and virile insistence on this innovation.

The crux of the opposition lay apparently in commercial reasons. It is true that some eminent engineers in civil life, who had no financial interests in the matter, were wholly sincere in voicing their disapproval of what, with their inadequate knowledge of naval conditions, seemed to them a hazardous venture. But, as to others, whether the basic reason for the war waged on the Bureau of Steam Engineering in opposition to the electric drive was patriotic or pecuniary may well be left to those who made the attack. I may say, however, that the question of royalties was involved in the difference between the use of the turbine in driving the propeller shaft directly or through interposed gearing, and its use indirectly as a primary source of power for generating an electric current to drive that shaft through an electric motor.

The conspicuous feature of the controversy thus forced on the Navy Department was the vociferous propaganda conducted by the opponents of the electric drive, in telegrams and letters to, and interviews with, the Secretary of the Navy; in a letter to the President of the United States; in letters to the chairmen of the Naval Committees of the Senate and the House of Representatives; and in a number of other ways, including a letter to the Secretary from a distinguished professor at a great University, recommending—at the request, he said, of somebody else—a proposition made previously that a Board of impartial experts be appointed by the Secretary of the Navy to consider and advise on the whole matter—that is, to tell the Bureau of Steam Engineering just what it should do in its own business.

And, too, this propaganda had its laughable side. For ex-

ample, one prominent electrical expert wrote vigorously against the use of electricity in this way, and then, apparently under business pressure, wrote again, and just as vigorously, approving its use—to the huge amusement of naval officials. And, further, the editor of a technical journal made himself absurd by presuming to tell naval officers and the Navy Department the kind of ships it should build, and by attempting to arrogate to himself a knowledge of naval matters superior to that of the General Board of the Navy, which had examined and approved the design he criticized.

Throughout this controversy the Engineer-in-Chief held unwaveringly to his decision. No arguments had power to turn him, for he had what his critics lacked, detailed knowledge both of naval requirements and of the performance of electricity as a driving power in naval vessels. About seven years before, the naval collier *Jupiter* had been thus equipped by his predecessor in office, Rear Admiral Cone. As soon as she was completed, Admiral Griffin, then Engineer-in-Chief, sent the Assistant to the Bureau—Commander, now Rear Admiral S. S. Robison—to the Mare Island Navy Yard, with orders to try the *Jupiter's* machinery at sea in every conceivable way. The report of this officer was exhaustive, and therefore Admiral Griffin knew in every detail just how the electric drive would act in all conditions. So from an engineering viewpoint, the contest between the Bureau and its critics was an unequal one.

However, that controversy matters little now, except, in its finish, as a warning to possible imitators of a future day; for the contract trials recently of the *New Mexico*—the first of our battleships to be fitted with the electric drive—have confirmed fully the judgment of the Engineer-in-Chief. She was tested in many more ways than her contract called for, and in all she has shown herself not only wholly efficient, but—in her system of propulsion, her maneuvering power, and her underwater protection, the leading battleship of her day in the navies of the world.

Now, let us turn to the technical side of this question. First, I need hardly say here that the steam turbine has replaced the reciprocating engine for battleship propulsion. And I need scarcely add that the combination of a turbine—whose maximum economy is shown at 2,000 revolutions or more—and a propeller—which, in a battleship, should not exceed 180 to 200 revolutions—presents a rotational problem of some complexity.

Evidently an intermediate reducing mechanism is required. There are now a number of these reduction gears—mechanical, hydraulic and electrical. The view shows the arrangement used in one of our older battleships. She is a twin-screw vessel, the two propeller shafts being shown at the left side of the diagram. The high-pressure and low-pressure turbines for each screw are in series and are mounted on separate shafts, each shaft carrying a pinion which engages in single reducing gear of large diameter keyed to the propeller shaft. Each low-pressure turbine shaft carries also an astern or backing turbine.

The principal objection to this arrangement is the necessity for, not only a backing turbine on each screw shaft, but one which gives only about 40 per cent of full power, since questions of weight and of space forbid the use of a backing turbine of the same size as the ahead turbine, and hence the result is a short one operating at low economy. A minor, but important, disadvantage is the very considerable width in a transverse or athwartship direction of the machinery space.

Now consider the electric drive, as shown diagrammatically by the view which follows. It will be seen that while the turbine is still the primary source of power, it operates an electric generator instead of a screw propeller, and, as the view of a four-screw ship shows, the electric power thus developed is transmitted to motors on the propeller shafts in much the same way as electric power is transmitted on shore. In other words, there is a central power plant—composed of

boilers, turbines and electric generators—and the power is taken from this plant to the motors on the screw shafts.

This installation possesses an advantage over the direct or geared-turbine drive, in that a backing turbine is not required. The reversing is done through the motors with the same ease and certainty as in any other electric motor installation, and with the further advantage that full power can be utilized in backing.

With a suitable construction of the motor, the turbines can be driven at maximum economy for both the full speed and the cruising speed of the ship; for other speeds, the turbine and alternator must be slowed down in the usual manner. This arrangement therefore gives a two-speed, reversible motor at maximum economy in both speeds and with full power in backing.

The elimination of the backing turbine is a matter of prime importance, since most of the accidents to turbine-driven vessels of the Navy are due to it. For example, the dreadnaught *Arizona*, in her first trip from New York, had been making a very fine record in straight-away steaming, but when she began maneuvering one of her turbines came to grief, and she had to limp back to the New York Navy Yard, with a big job of re-blading to be done.

The electric drive has numerous other advantages, which I cannot now recount. In sum, its use gives:

Greater flexibility and control in the use of power.

Improved economy in every-day service and good economy at high speeds.

Less liability to serious derangement of machinery.

Less likelihood of the speed of the ship being seriously affected in the event of injury to the turbines.

And, these two paramount advantages:

Military superiority from greater maneuvering power;

Military superiority due to increased under-water protection.

In the noble future which the destiny of nations foreshadows for this Republic two things seem clear to our vision now: first, that the United States is to have a huge merchant marine, perhaps the largest in the world; second, that for guarding that commercial fleet and to do our share in keeping world-peace we are to have also a Navy at least equal to that of any other nation. So any advance which adds to the efficiency of our fleet is of fundamental value to us.

This the electric drive does. Its advent has, in fact, revolutionized the propelling systems of the battleships of all navies. Secretary Daniels has written, in courtly phrase, of "the vision and the wisdom of Rear Admiral Griffin" in foreseeing all this; but it should not be forgotten that the Secretary of the Navy, to his everlasting honor, stood loyally by the Engineer-in-Chief throughout this bitter controversy. And, too, we owe tribute to a distinguished engineer, Mr. W. L. R. Emmett, an Annapolis graduate, for his steadfast advocacy of the electric drive, and, as well, to the great electrical organization on whose staff he is, and whose enterprise and business sagacity made this advance possible.

## CONCLUSION.

I have endeavored this evening to give in swift survey some account of the achievements of the Bureau of Steam Engineering during this war. The scope and magnitude of its work have been so great, however, that my resumé has been necessarily both general and incomplete. And yet, one fact should be clear: that an immense amount of work has been done, very rapidly and with full efficiency.

As to that work I may say that, while I have no authority to speak for the Engineer-in-Chief, I am sure that I voice his views when I state—and with pleasure before this Society—that, although all of this work has been done under direction of the Bureau, it could not have been accomplished without the hearty coöperation and assistance of those ship and engine-

building establishments which delivered battleships, destroyers, submarines and submarine chasers to the Navy during the war.

Still less would this work have been possible without the whole-hearted support of the large manufacturers of steel, and of copper and brass, who were the first to place their products at the disposal of the Government. To the great electric companies, to a large manufacturer of boilers and another of pumps, and to the smaller manufacturers, whose name is legion and whose output is to be found on every ship, great credit is due for their aid in making successful so stupendous an undertaking.

And, last, but far from least, our Navy Yards and the many private plants whose facilities have been utilized in repairing the machinery of our ships, and in thus maintaining the efficiency not only of the Fleet but of the vessels engaged in the transport of troops and munitions, deserve the highest praise for the energy and efficiency which has characterized their work during this war.

I have added pleasure in this tribute here, since so many of the men interested in this vast aggregation of engineering and shipbuilding enterprises are members of this great national Society.

# DYNAMICS OF GEAR DRIVE.

By N. W. Akimoff.

In a recent remark in the "International Marine Engineering" I gave a short outline of the problem of the rational design of the gear drive.

There is absolutely no escape from the fact that, so far, the design of this drive has been based only on the *Static* considerations; the *Dynamics* has always been left severely alone, assuming therefore—

1. That the shaft is *infinitely short*, and just as *infinitely rigid*.

2. That neither the gear nor the propeller weigh anything at all, or that their radii of gyration are next to zero.

3. That, instead of the propeller we have something like a generator, under an absolutely constant load.

I will ask the reader, how reasonable these assumptions really look. The whole thing boils down to this: *do we want to know what we are doing or don't we?* Here are the facts:

1. The shaft is generally both long and comparatively slender.

2. Both the gear and the propeller are exceedingly heavy, and their moments of inertia are tremendous.

3. The resistance offered by the propeller is not constant but periodic, each blade contributing one maximum and one minimum during each revolution; so that, for instance, for a three-bladed propeller we have three impulses, of some kind, per revolution. Just what these impulses are is another matter, not in the least interesting from the immediate standpoint of the present discussion. Prof. G. K. Calhoun, U. S. N., thinks that the predominant cause of these variations lies in the proximity of the rigid boundary (the hull proper); while, on the other hand, I personally feel that the variation is

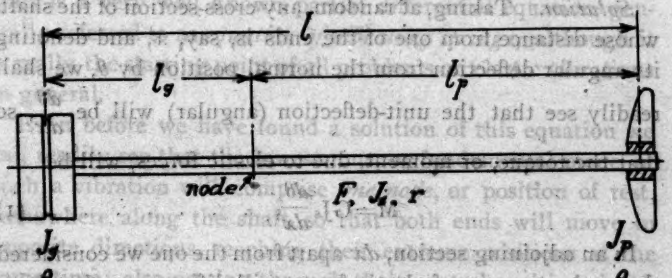
largely due to the variable submergence of the revolving blade, wrapped up in its vortex, all of which will be submitted later, in a separate communication. From Admiral Dyson's remark on page 107 of his book on Propellers, it appears that tip clearance has a great deal to do with vibrations in general.

At any rate these three (or whatever the number might be) impulses per revolution, are somewhat in the nature of external periodic force (we say *force*, in the generalized sense, of course), the magnitude of which we neither do nor care to know, but whose frequency, at least, is something tangible.

They are probably not very great in themselves, but if their frequency is the same as (or near) that of one of the natural modes of the angular, torsional, vibration of the elastic dynamical system, *gear-plus-shaft-plus-propeller*, then synchronism will take place and there will be trouble.

So, if we really mean to know what we are doing we must either make sure that such synchronism will not take place; or, if it cannot be avoided, for some reasons, that we have at our disposal certain means whereby to paralyze its effect.

The most important part of the problem, therefore, is to know at least the fundamental frequency (the gravest mode) of the system shown in the sketch.




The fact that the shaft does not float in the air but is mounted in bearings, does not matter very much; friction, unless abnormal, does not materially change the rate of free vibrations.

That such torsional vibrations actually take place has been known for the past eighteen years. Dr. Hort's beautiful book, *Technische Schwingungslehre*, gives a fairly complete bibliography of this subject; in more recent years the matter of torsional vibrations has been looked into, from experimental standpoint by our Navy Department, and the article on the subject, by Naval Constructor Wm. McEntee, published in this Journal, is no doubt well familiar to its readers.

However, most of these papers and articles deal with engine drive, assuming the resistance of the propeller to be constant and not periodic, and dealing with considerable amplitudes.

We shall therefore derive a complete solution of the problem in a manner that could be adapted to be of immediate benefit to the designer of a gear drive:

*Problem.* The length of the shaft is $l$, the polar moment of inertia of its cross-section is $J_s$; the area of cross-section is $F$; the modulus of elasticity (shear) is $G$; and the mass of one linear foot of shaft is $\gamma$, the radius of gyration of the cross-section being $r$. The weights of the gear and the propeller are $W_g$ and $W_p$; their moments of inertia are $J_g$ and $J_p$. Everything in feet and pounds. *Find the frequency of the fundamental mode of the torsional vibration of such a system.*

*Solution.* Taking, at random, any cross-section of the shaft, whose distance from one of the ends is, say, $x$, and denoting its angular deflection from the normal position by $\theta$, we shall readily see that the unit-deflection (angular) will be $\frac{d\theta}{dx}$, so that the torque, or moment, due to elastic forces, will be

$$M = GJ_s \frac{d\theta}{dx}, \quad \ldots \ldots \ldots \quad (1)$$

In an adjoining section, $dx$ apart from the one we considered above, the moment of elastic forces will differ somewhat from the above; it will be

$$M + dM = GJ_s\left(\frac{d\theta}{dx} + \frac{d^2\theta}{dx^2}dx\right) \quad (2)$$

(by Taylor's series).

Hence an immediate solution of the problem of the motion (torsional) of the slice $dx$ itself: indeed, the moving or external force (in our case *moment*) will then be

$$dM = GJ_s \frac{d^2\theta}{dx^2} dx, \quad \ldots \ldots \quad (3)$$

while the *mass* (rotational) will be simply the moment of inertia of the slice $dx$, that is

$$\gamma r^2 dx;$$

the *acceleration* of this element (of course *angular*) will be $\frac{d^2\theta}{dt^2}$ and since the principle *force* = *mass* × *acceleration* applies to rotational motion, if only instead of *force* we take the corresponding *moment*; instead of *mass* the *moment of inertia*, and, instead of linear *angular* acceleration, we have

$$GJ_s \frac{d^2\theta}{dx^2} = \gamma r^2 \frac{d^2\theta}{dt^2}. \quad \ldots \ldots \quad (4)$$

Dividing by $J_s$, which of course is $= Fr^2$, we have

$$\frac{GF}{\gamma} \frac{d^2\theta}{dx^2} = \frac{d^2\theta}{dt^2};$$

or, representing, for convenience, $\frac{GF}{\gamma}$ by $a^2$, we finally have

$$\frac{d^2\theta}{dt^2} = a^2 \frac{d^2\theta}{dx^2}, \quad \ldots \ldots \quad (5)$$

This is a very well known partial differential equation, generally referred to as *equation of vibrating strings;* although it is really the starting point of all problems on elastic vibrations in general.

Even before we have found a solution of this equation we can readily see that the lowest tone, or fundamental mode, of such a vibration will comprise *one node*, or position of rest, somewhere along the shaft, so that both ends will move in opposite directions, reaching their extreme positions at the same time; also passing through their central position simultaneously.

For an unloaded shaft of uniform cross-section the node will naturally be in the middle; for a shaft, having heavy masses on its ends, nothing can be stated, beforehand, regard-

ing the position of the node, except that there will be one, somewhere.

The particular solution of the equation (5) is sought in the form

$$\theta = A.\ \sin pt.\ \sin mx. \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Here the constant A has something to do with the actual amplitude, and from our standpoint is only of secondary interest. But the constants $p$ and $m$ are of immediate importance: $p$ is the frequency-constant and $m$ apparently has something to do with the form of the curve (sine curve in this case) which will be the diagram of the angular twist (*not* the elastic curve, as would be the case in a problem on transverse vibrations).

In order to see if there is any connection between $p$ and $m$, such as would enable them both to satisfy our tentative solution (6), let us substitute the latter into (5), upon proper differentiation as to $x$ and $t$. We have, upon obvious reduction

$$p^2 = a^2 m^2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

which establishes the relation between $p$ and $m$.

We shall now look into the *end conditions:* let us assume that the location of the node is established by $l_g$ and $l_p$, both lengths so far unknown.

Consequently the torsional moment on the left end will be found by integrating (3) between zero and $l_g$; this will give

$$M_g = GJ_s \left(\frac{d\theta_1}{dx}\right)_{x=l_g}$$

This very moment is operating on the gear, whose moment of inertia is $J_g$; applying therefore the same principle (and in the same sense) as above, we have *force = mass × acceleration*

$$GJ_s \left(\frac{d\theta_1}{dx}\right)_{x=l_g} = -J_g \frac{d^2\theta_1}{dt^2}. \quad . \quad . \quad . \quad (8a)$$

and, in the same manner, for the other end

$$GJ_s \left(\frac{d\theta_2}{dx}\right)_{x=l_p} = -J_p \frac{d^2\theta_2}{dt^2}. \quad . \quad . \quad . \quad (8b)$$

The sign — means that the moment (in our case the elastic resistance) tends to bring the body toward zero position (of

rest), as is the case in all elastic vibrations. Utilizing the form (6) adopted above, we have

$$GJ_s \sin. pt. \cos. ml_g = J_g p^2 \sin. pt. \sin. ml_g, \quad . \quad . \quad (9)$$

$$GJ_s \sin. pt. \cos. ml_p = J_p p^2 \sin. pt. \sin. ml_p,$$

or, simplifying and remembering that

$$p^2 = a^2 m^2, \text{ also that } a^2 = \frac{GF}{\frac{l}{3}},$$

we have these two equations in their final form

$$m \tan (m.l_g) = \frac{J_1}{J_g} \quad . \quad . \quad . \quad . \quad (10)$$

$$m \tan (m.l_p) = \frac{J_1}{J_p}$$

where $J_1$ is, of course, the moment of inertia of one linear foot of the shaft about its axis.

To these we must add the obvious relation

$$l_g + l_p = l, \quad . \quad . \quad . \quad . \quad . \quad (11)$$

and these three equations will determine the three unknowns of our problem, namely $l_g$, $l_p$ and $m$.

These are found by the method of successive approximations. First, we assume a certain value of $l_g$ and find the corresponding value of $m$ from the first equation (10); this value of $m$ will now be substituted into the second equation (10) and the result solved for $l_p$. If the value of $l_p$ thus found does not check up with the condition (11), another trial should be resorted to.

According to the remark made by Prof. Timoshenko (of St. Petersburg) in his paper on shaft vibrations, we should always begin by the following approximate value of $l_g$

$$l_g = l \frac{J_p + \frac{lJ_1}{3} \cdot \frac{J_p}{J_g + J_p}}{J_g + J_p + \frac{lJ_1}{3}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Having found a satisfactory value of $m$ we must utilize it for finding the desired frequency per minute, N.

The following reasoning will be found quite obvious since $p^2 = a^2 m^2 = \frac{GF}{\gamma} m^2$, we have

$$p = \sqrt{\frac{GF}{\gamma}}\, m = \frac{2\pi}{60} N,$$

so that

$$N = \frac{30}{\pi}\sqrt{\frac{G}{\gamma}}\, m \sqrt{F} = 103{,}000\, m \sqrt{F} \quad . \quad . \quad (13)$$

The reader, if interested, will check the following numerical example:

$J_1 = 23.2$; $J_g = 12{,}500$; $J_p = 3{,}550$; $F = .8$ sq. ft. $l = 122$ ft.

Here $m$ comes out to be $= .0077$.

The position of the node is given by

$$l_g = 31 \text{ ft}; \quad l_p = 91 \text{ ft};$$

and, finally, the number of free torsional vibrations per minute will be

$$N = 103{,}000 \times .0077 \sqrt{.8} = \text{about } 710.$$

This means that, for a three-bladed propeller, the speed of 236 R.P.M. will lead to synchronism—and therefore to trouble.

In solving equations (10) and (11) a 20 inch slide rule can be used to advantage; also a table of natural values of circular functions similar to that given on page 20–22 of Hoüel's *Recueil de formules.* (Paris, 1901).

# ELECTRIC WELDING PLACED ON A SCIENTIFIC BASIS.

By Captain of Engineers J. H. Chalker, U. S. C. G., Member.

In order to insure a perfect weld the elements which constitute the electrode and the object to be welded must be analogous, and this can only be determined by research and experiment. For instance, take two pieces of steel from the same plate and weld them; then place the welded piece in a testing machine and pull until ruptured; if the steel is found to have, for instance, a tensile strength of 60,000 pounds and an elongation of, say 25 per cent in eight inches, this steel upon analysis would probably show about .162 to .182 carbon and about .34 manganese, but to be certain, the steel constituting the test piece as well as the electrode should be analyzed.

It is a well known fact that with ordinary open-hearth steel ship and boiler plates the amount of carbon and manganese are the deciding factors governing the tensile strength and elongation of the material, although both tensile strength and elongation can be raised or lowered by cold or hot rolling.

Now continue the experiments with testing machine and analyses of steel of various tensile strengths and elongations, electrodes of known elements and an electric current of known voltage, and amperage.

The carbon and manganese contained in electrode should bear a certain relation, depending upon the strength of the weld, to the carbon and manganese in the metal to be welded; that is, the per cent of carbon and manganese in the electrode should be high enough to allow a certain portion of both to be burned out in the process of making the weld and still contain the necessary amount to insure a strong weld. For instance,

it might be necessary to use an electrode containing .29 per cent of carbon and .60 per cent, or upward, of manganese, while the weld when completed might contain but .17 per cent carbon and .32 per cent manganese. If all the carbon and manganese contained in the electrode were burned out the tensile strength of the weld would probably not be greater than that of wrought-iron while it should approach the tensile strength of the material welded.

After a series of tests and analyses together with currents of various voltages and ampères a point is arrived at where a weld ceases to be, so to speak, an experiment in any sense of the word, the only undetermined factor being the thickness or bulk of plate to be welded. The welder is no longer obliged, except in isolated cases, to have the material which is to be welded analyzed. It is only necessary to refer to the table which has been compiled to know which electrode, voltage and number of ampères to use for steel of known chemical constituents. In other words, knowing the constituents of the metal to be welded one would simply use an electrode and current, determined by these experiments, best adapted to assure a perfect weld.

Referring to the electric welding of certain parts of the steam machinery of German ships under the supervision of the Navy Department, the work was performed under circumstances similar to those outlined in this paper; that is, a constant potential circuit was used, the same being not over 40 volts open circuit. The current was regulated for a predetermined value and automatically controlled so as not to vary more than 5 per cent either way.

Of course it is realized that the skill of the welder will always be a dominating factor in good welding, but this might be termed a commercialism and a case of "you pay your money and you take your choice," for the best welders should command the highest compensation.

It is only necessary to know the heat or melt number, which

is stamped on each ship and boiler plate, to obtain from the manufacturer both the physical and chemical tests, and as a matter of fact this data is usually required by ship builders with the shipment of plates from the mill.

The same experiments should be made with cast iron and other metals to be welded, and thus place electric welding on a secure and scientific basis and eliminate all guessing.

These experiments could be conducted by the Bureau of Standards, with various welding machines or perhaps the better procedure would be for the manufacturer of each type of welding machine to conduct his own experiments together with his own analyses, and furnish the necessary data with each machine placed on the market.

The compilation of data would not be either a long or an expensive task, for the range of physical and chemical requirements for steel plates, shapes and castings are not great, say between 60,000 and 70,000 pounds tensile strength, and as before stated the amount of carbon and manganese is in direct proportion to the tensile strength and elongation, the carbon being the governing factor. The higher the tensile strength of the steel the less will be the elongation.

Phosphorous and sulphur should be as low as possible, but in open-hearth steel made under the basic process neither the phosphorous or sulphur is usually high.

# OPTICAL METHOD OF SHAFT ALIGNMENT.

## BEING A REPORT OF REALIGNMENT OF INBOARD PROPELLER SHAFTS U. S. S. "MISSISSIPPI," NORFOLK NAVY YARD, JUNE, 1918.

By Captain William Norris, U. S. Navy, Member.

The U. S. S. *Mississippi* had reported excessive vibrations, under way, in the vicinity of the starboard inboard propeller, with leakage around strut palms and hull plate seams. As the spring bearings of No. 2 shaft had occasionally given trouble, it was believed desirable to check alignment of this shaft at the first opportunity.

The *Mississippi* was docked, June, 1918, the starboard inboard propeller and stern-tube shafts removed, tested in the shop, and found true within .010". A line, the usual wire, was centered in the inboard and outboard stern-tube bearings and carried through the strut. The strut appeared to be so far out of line that it was decided to recheck, using the optical method recently developed at this Yard (described later). Measurements taken checked very closely with the ordinary line, the measurements showing the center of the bore of the struct 7/16" low and 3/16" outboard, in the docked condition, with respect to the centers of the stern-tube bearings.

The port inboard propeller and stern-tube shafts were then removed, and alignment checked as in the case of the starboard inboard shafting, the center of the strut in this case being down 9/32" and outboard 1/8".

It was thought that this condition might be caused by docking, as the ship had been on the blocks for several days without the overhang of the stern being supported from the dock floor. At this time, other urgent docking work required that the *Mississippi* be undocked, and, during the undocking, the

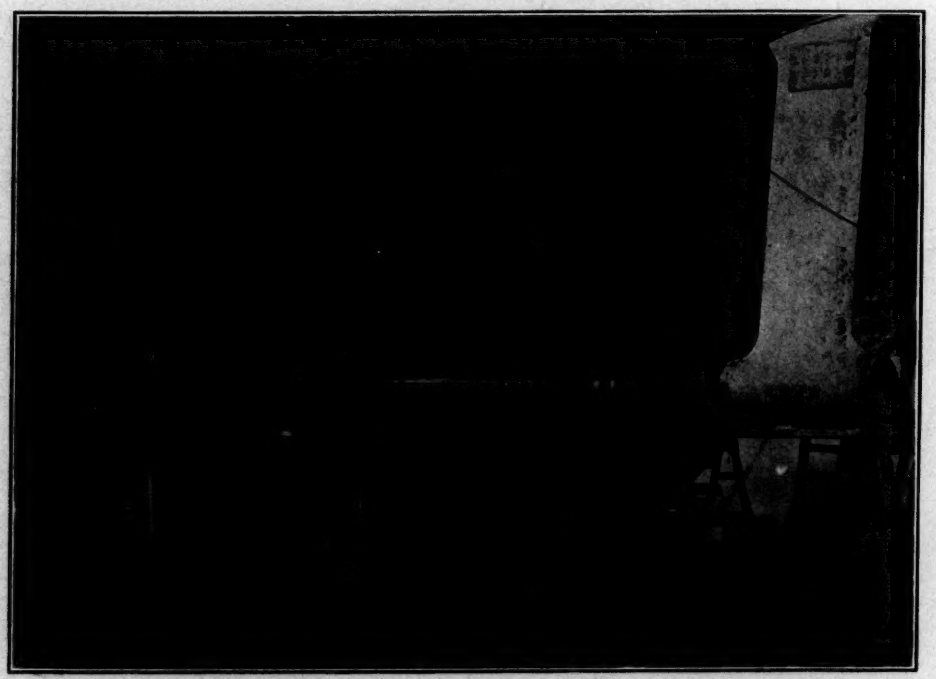


PLATE I.

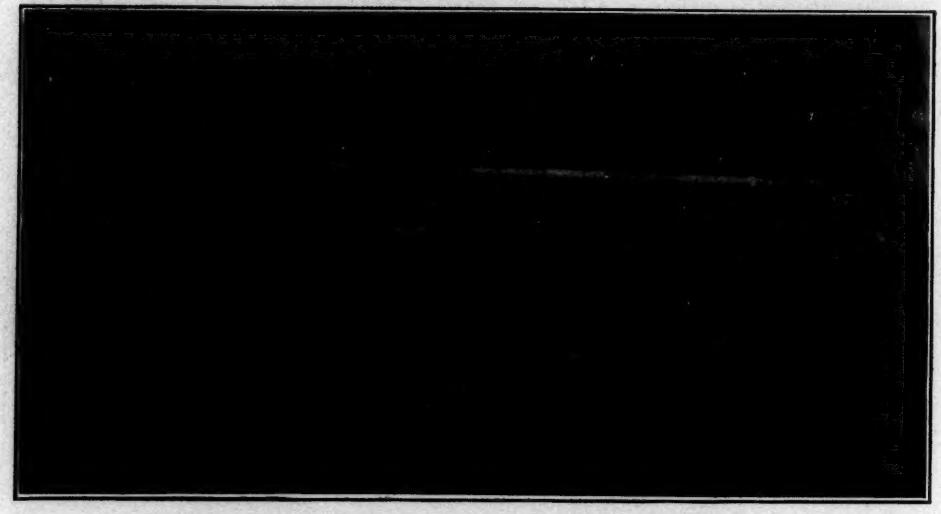
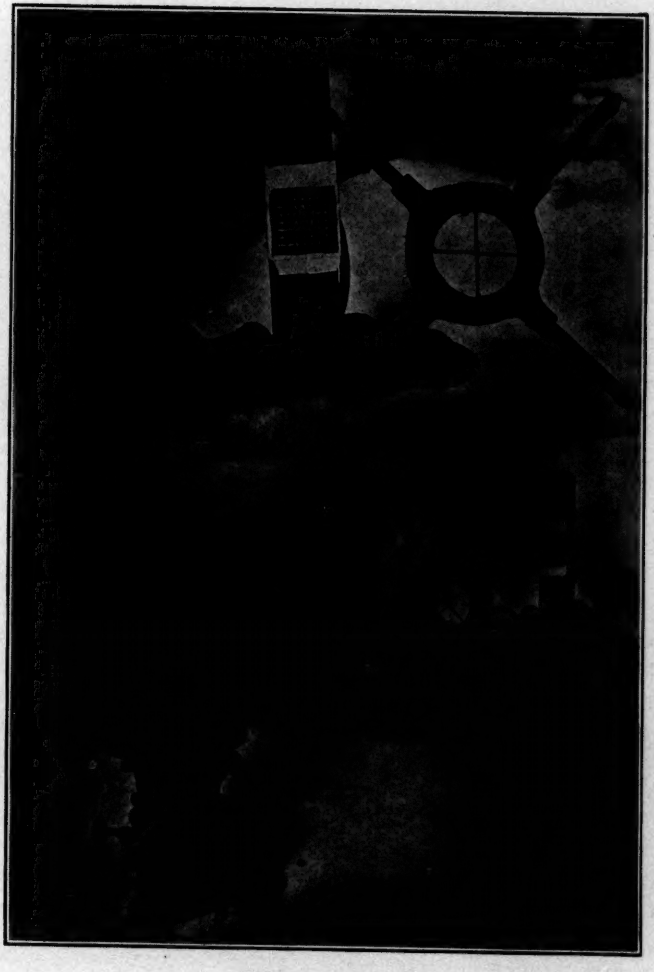
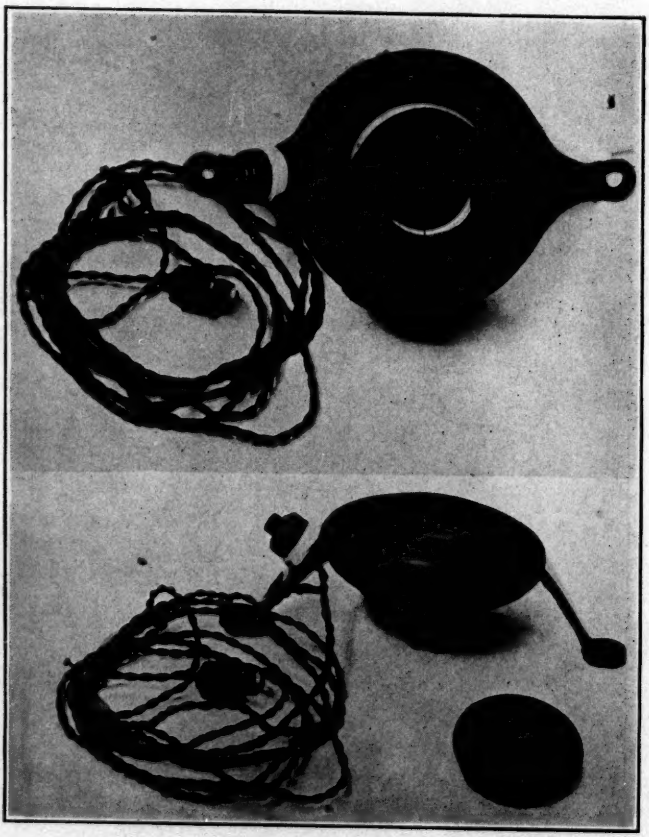


PLATE 2.



PLATES 3 AND 4.



PLATES 5 AND 6.

opportunity was taken to determine whether or not the stern would rise or fall relative to the structure as a whole when the vessel was water borne. The result of this investigation indicated an actual sagging of the stern when the vessel floated. Measurements of this deflection were taken with a surveyor's level mounted on deck close to the after turret, readings being taken before the dock was flooded and with the vessel afloat.

Two facts were thus determined: first, as docked, the struts for Nos. 2 and 3 shafts were considerably out of line with reference to the stern-tube bearings; secondly, there were indications of sagging of the stern relative to the hull as it had rested on the blocks when the vessel was floated. It was thus necessary to determine the alignment of the strut with respect to the stern-tube bearings *with the vessel afloat;* and the method of determining this is described as follows:

In general, the scheme was to project a beam of light from a target located in the strut through the stern-tube to a surveyor's transit located in the shaft alley, a 6″ watertight tube (see Plates Nos. 1 and 2) being fitted between the forward face of the strut hub and the outboard stern-tube bearing, the after face of the strut hub being blanked off. Spiders carrying perforated centers were located at the exact center of bore of both inboard and outboard stern-tube bearings. Then by locating a transit so that the cross-hairs were in line with the centers of the stern-tube bearings, a line of sight was established which, when projected on to the target in the strut, showed directly the deflection of the center of the strut with reference to the stern-tube bearings.

Plates 1 and 2 show the watertight tube from strut to stern tube and the method of securing, also the flexible flanged joint to take up any changes in the hull structure, allow for expansion and any deflection that might be due to buoyancy of the tube. As a precaution in case the tube should carry away, the inboard end of the stern tube was also blanked off, a hole being

left to lead the electric-light wires through to the target and a three-inch hole through which to sight the transit, the latter hole being tapped and a pipe plug, and also a wooden plug, kept close at hand for use in emergency. After the tube had been rigged and before the dock was flooded, air pressure was put on the stern tube, connecting tube and struts to test for leaks and several holes were found in the hub of the strut, which were plugged. The target (Piece T, Plates 3 and 4) was secured in the strut before the connecting tube was secured in place. The target consists of a 3-inch diameter brass tube 12 inches long, mounted on a piece of brass plate, and carrying on the inside of it two electric lights (carbon). A 2½-inch square section is cut out of this tube, and the target proper is fastened over the opening. The target consists of a framework of brass with 1/32 inch holes drilled on 1/16-inch centers around the sides of the frame and waxed linen thread laced through these holes as shown on Plate 3. The particular system of coördinates used is to facilitate the reading of the exact point where the cross-hairs of the transit strike the target. The center of the target, the intersection of the two diagonal threads, was located at the center of the bore of the strut hub. The electric light leads ran through the tube and stern tube. The centering spiders (Pieces S, Plates 3 and 4) were located in the inboard and outboard stern-tube bearings with their centers coincident with the center of bore of the bearings. These two spiders formed locating points for establishing the line of sight.

After locating the spiders in the stern tube and the target in the strut hub, the expansion tube and blanks were bolted in place, the air test for tightness made, and the ship floated.

The transit was adjusted approximately at any convenient time after the stern-tube spiders were installed and secured, and checked after the vessel was afloat. A special base plate (see Piece B, Plates 3 and 4) was found very convenient for quickly locating the transit.

With the above equipment in place and the light turned on the target, it was a simple matter to set the transit in line with the centers of the spiders in the stern tube by alternately focusing the transit on each center. This established the line of sight, and it was then only necessary to focus the transit on the target. The point where the cross-hairs of the transit intersected the target indicated the true center line of the stern tube extended. The amount this point differed from the intersection of the diagonal threads on the target was the amount the center of the strut hub was out of line with the center line of the stern tube, and this amount could be read directly from the target. In applying the necessary correction of misalignment, its direction will be determined by the type of transit used, whether inverting or erecting.

Plate 4 shows the sequence of setting up the various parts, and also the necessity of cutting away the caps of the spiders to allow focusing through the first spider to the second spider and from there to the target.

The usual precautions for handling a precision instrument must be observed in setting and focusing the transit, but with reasonable care and an accurate instrument, the maximum error should not exceed 1/32 of an inch at 125 feet, with a transit magnifying 24 diameters.

Measurements taken on the *Mississippi, the vessel afloat*, with the apparatus described above, showed that the center of the port inboard propeller strut hub was 13/64″ outboard and 3/8″ lower than the center line through the stern-tube bearings; and that the center of the inboard starboard propeller hub was 3/16″ outboard and 3/8″ low.

Readings of this misalignment were taken both with the ship on the blocks before flooding and immediately after flooding, and it was observed that the sagging of the stern upon floatation of the ship was not more than 1/16 of an inch. Plate 7 shows blue prints of the targets in these instances, the dotted

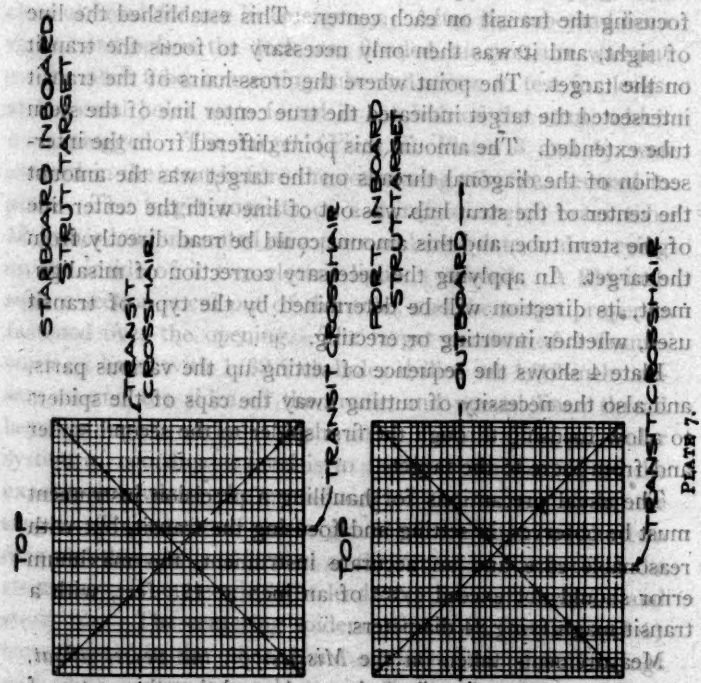

STARBOARD INBOARD STRUT TARGET
TRANSIT CROSSHAIR
TOP
OUTBOARD
PORT INBOARD STRUT TARGET
TRANSIT CROSSHAIR
TOP
OUTBOARD
TRANSIT CROSSHAIR

PLATE 7.

line indicating the location of the transit cross-hairs on the target with the ship afloat.

The errors noted were corrected by reboring struts and fitting new strut-bearing bushings.

In conclusion, it is stated that this method of optical alignment has been successfully used at this Yard for establishing the shaft lines on vessels under construction. If the coördinates of any two points along the shaft line are given—for instance, a point on a bulkhead and the location of the center of the stern-tube bearings—the location of all the other bearings along the shaft line can be determined, it being only necessary to set up the transit in line with the two given points, and, without changing the setting of the transit, focus it on any other point in line and set the centers of the spiders coincident with the cross-hairs of the transit. The center of the spiders then form a locating center from which to lay off points preliminary to boring.

Instead of using the target which is described above, when establishing a shaft center line, a special target (see Plate 5) or light-cross is used. As will be seen from Plates 5 and 6, this target is a bracket-like device on the back of which is mounted a parabolic reflector carrying a low candlepower electric light and on the front a cap. Through this cap are cut two 1/32″ slots forming a cross through which the light from the reflector is projected onto the lenses of the transit. This target is mounted on a bulkhead with the center of the light-cross as one established point in the center line of the shaft. The cross-hairs of the transit may be very accurately located in the light-cross and the reflected light from the target (with the cap removed) allows of the work being carried on at night when the hull is of a uniform temperature and interference by other workmen on the ship is eliminated.

The shaft lines of Destroyer No. 70, *Craven*, were thus located, the time required for both shafts being about three hours. The alignment was later found to be very good.

**The details and general arrangement were perfected by Mr. L. C. Campbell, Junior Inspector of Naval Construction, to whose interest and initiative the successful development of this method of shaft alignment is largely due.**

# AN IMPROVED METHOD OF OPERATING EVAPORATORS.

By M. C. Stuart,* Associate.

Marine Engineers will agree that the evaporating plant has the unenviable reputation of requiring more attention and causing more trouble than any other of the ship's auxiliaries. The determination of the proper coil pressure, water level and other conditions for the production of a desired capacity forms a source of continual speculation. Priming, reduction in capacity and the rapid accumulation of heavy masses of scale are the accompaniments of the most careful operation. Recent improvements in design of shell and coil leave untouched the many problems and difficulties of operation.

A new method of operating evaporators, developed at the U. S. Naval Engineering Experiment Station, Annapolis, Md., shows such remarkable and positive results in solving the questions and lessening the difficulties of operation as to warrant its careful consideration, and, it is believed, its adoption, by Engineer officers who are interested in maintaining the ship's fresh-water supply with the least possible effort.

The essential feature of the method is the production of fresh water at a constant rate and at any desired capacity within the limit of capacity of the evaporator. This result is obtained by the application of a steam orifice to the coils, which, by reason of the law of constant flow of steam, as expressed by Napier's formula, effects a constant production of vapor without the need of additional regulating appliances. In addition to the constant capacity feature, the use of the method reduces to a minimum the liability of priming, con-

*Mechanical Engineer, U. S. Naval Engineering Experiment Station, Annapolis, Md.

siderably lessens the difficulty with scale, and stabilizes and simplifies the entire operation.

Before describing the method it will be desirable to review the process of evaporation, and discuss at some length the usual method of operation, in order that the principle of the new method and the manner in which it remedies some of the difficulties and defects of the usual method may be clearly understood.

The production of fresh water from sea water is accomplished by the vaporization of sea water by means of the condensation of steam in coils which are surrounded by the sea water or brine contained in a shell. The condensed steam is drained from the coils and the vapor is condensed in a separate distiller condenser, usually called the distiller. In a multiple-effect plant the vapor produced is carried along to serve as steam in the coils of a second evaporator or effect. As evaporation continues, the water level of the brine in the shell must be maintained by the continuous introduction of sea water as feed. This increases the amount of salt in the brine and the accumulated salt must be removed, either by the continuous discharge of a small amount of brine (continuous blow-down), or by the periodic discharge of the entire volume of brine when the salinity reaches a certain limiting value (intermittent blow-down).*

## HEAT TRANSFER COEFFICIENT.

An unavoidable result of the process of evaporation is the gradual accumulation of scale on the brine side of the coils. This scale greatly affects the ability of the coil to transmit heat, and is one of the most troublesome features of evaporator operation. The ability of the coil to transmit heat is measured by the heat transfer coefficient, and inasmuch as the entire process of evaporation is so closely bound up in the

*For a complete discussion of blow-down, see "Salt Water Evaporators," by W. L. De Baufre, A. S. N. E. JOURNAL, Nov., 1915, page 946.

transmission of heat through the coil the heat transfer coefficient may be considered at some length with profit.

By definition, the heat transfer coefficient is equal to the B.t.u.'s transferred per hour per square foot of surface per degree temperature difference. This may be expressed by the formula,

$$K = \frac{W\,(H_s - q_d)}{S \times T.\,D.},$$

in which,

K = Heat transfer coefficient.

W = Steam condensed in coils, pounds per hour.

$H_s$ = Heat in steam entering coils, B.t.u. per pound.

$q_d$ = Heat in condensate from coils, B.t.u. per pound.

S = Evaporating surface, square feet, measured on the outside of the coils.

T. D. = $T_s - T_v$, in which, $T_s$ is the temperature corresponding to the pressure of the steam in the coils, and $T_v$ is the temperature of the vapor corresponding to the vapor pressure in the shell.

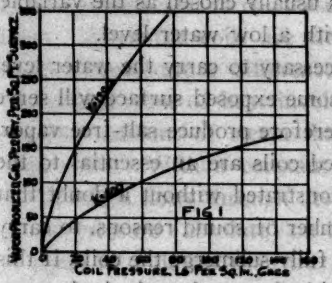
The numerical value of the coefficient for any given set of operating conditions may be computed from the equation. Physically, the value of the coefficient depends upon a number of factors, which, in the probable order of their importance, are condition and amount of scale accumulated, salinity of brine, water level, design of coils, vapor pressure and coil pressure. In a given evaporator, values of K ranging from 1,400 to 100 are possible. One thousand two hundred is an average value when the coils are clean and with sea water in the shell, and 200 is a probable average value when the scale has accumulated on the coils to a considerable extent.

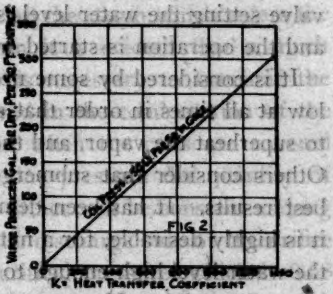
In the practical operation of an evaporator by the usual method a most important problem is the determination of the proper coil pressure for the production of a desired capacity. The temperature difference required for the production of any

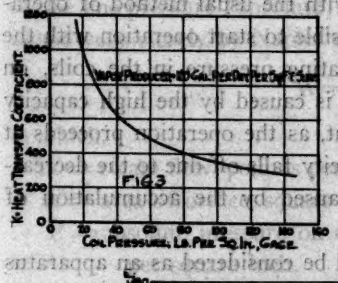
capacity may be determined at any time from the heat transfer coefficient equation if the value of K be known. Having the temperature difference, the coil pressure corresponding to the temperature difference is easily determined. Figure 1 shows the relation between coil pressure and capacity for a single-effect plant, with atmospheric shell pressure and values of K = 1,200 and 400. The curve for K = 1,200 will apply approximately when the coils are clean and scale free. As scale forms on the coils the value of K decreases, and at a constant coil pressure the capacity will decrease as shown by the curve of figure 2. This immediately suggests that in order to maintain the capacity as the value of K decreases it will be necessary to increase the coil pressure. The relation between heat-transfer coefficient and coil pressure necessary to produce a given constant capacity is shown by the curve of figure 3. The procedure of continually changing the coil pressure would be very troublesome and is seldom used. What is usually done is this: A coil pressure is selected which will give the capacity desired when the coils are partially covered with scale. This pressure is then maintained on the coils at all times by means of a reducing valve in the steam line to the coils.*

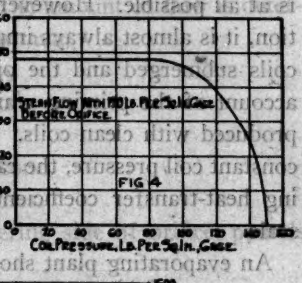
When the coils are clean the capacity produced is considerably above the normal amount, and in most evaporators the starting must be made very cautiously on account of the danger of priming, which is a natural result of high capacity. In order to minimize the danger of priming it is a usual practice to carry either a lower pressure or a lower brine level at starting. On account of the trouble of changing the reducing-

*The question of proper coil pressure is treated by Comdr. H. C. Dinger in an article, "Notes on Evaporating Plants," A. S. N. E. JOURNAL, August, 1914, page 849, as follows: "When the coils are clean a much lower pressure will suffice than when heavy with scale. As coils become dirty increased pressure will have to be used to secure the same capacity. It is, however, deemed inadvisable to be continuously changing the reducing-valve pressure. It would appear better to keep a certain pressure, which, in the double-effect plants of the Navy, should be about 60 pounds, and run with this all the time. When the capacity is materially reduced by salt deposit the evaporator should be scaled."



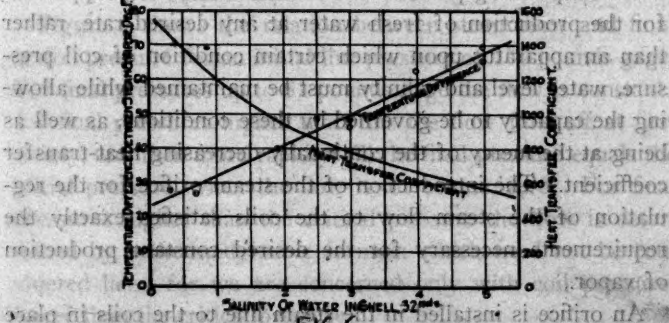




FIG. 1.—COIL PRESSURE AND CAPACITY.

FIG. 2.—HEAT TRANSFER COEFFICIENT AND CAPACITY.

FIG. 3.—COIL PRESSURE AND HEAT TRANSFER COEFFICIENT.

FIG. 4.—STEAM FLOW THROUGH ORIFICE.

FIG. 6.—EFFECT OF SALINITY OF BRINE.

valve setting the water level is usually chosen as the variable, and the operation is started with a low water level.

It is considered by some necessary to carry the water level low at all times in order that some exposed surface will serve to superheat the vapor, and therefore produce salt-free vapor. Others consider that submerged coils are an essential to the best results. It has been demonstrated without a doubt that it is highly desirable, for a number of sound reasons, to carry the water level high enough to fully submerge the coils, if this is at all possible. However, with the usual method of operation, it is almost always impossible to start operation with the coils submerged and the operating pressure in the coils, on account of the priming which is caused by the high capacity produced with clean coils. But, as the operation proceeds at constant coil pressure, the capacity falls off due to the decreasing heat-transfer coefficient caused by the accumulation of scale.

An evaporating plant should be considered as an apparatus for the production of fresh water at any desired rate, rather than an apparatus upon which certain condition of coil pressure, water level and salinity must be maintained while allowing the capacity to be governed by these conditions, as well as being at the mercy of the continually decreasing heat-transfer coefficient. The introduction of the steam orifice for the regulation of the steam flow to the coils satisfies exactly the requirements necessary for the desired constant production of vapor.

An orifice is installed in the steam line to the coils in place of the usual reducing valve. Full main-line pressure is carried on the inlet side of the orifice, and the orifice discharges directly to the coils. The steam which flows through the orifice is condensed in the coils, and only condensed steam is allowed to discharge from the coils, the use of a drain pot or trap insuring this. The pressure in the coils automatically adjusts itself to a value which will produce a temperature

difference sufficient to condense all the steam which enters the coils. When the coils are clean it is evident that a much smaller temperature difference is necessary to transmit the constant amount of heat involved in the condensation of the constant amount of steam than when the coils are covered with scale. With a constant line pressure the quantity of steam which flows through the orifice will be constant as long as the pressure (absolute) on the discharge side of the orifice is equal to or lower than the critical pressure, which is 0.58 of the line pressure (absolute). The amount of flow is determined very closely by Napier's formula,

$$W = \frac{P A}{79},$$

in which,

W = steam flow, pounds per second.

A = area of orifice, square inches.

P = absolute pressure on entering side of orifice, pounds per square inch.

Figure 4 shows the relation between steam flow and final or coil pressure for an initial line pressure of 150 pounds gage. For final pressures up to 81 pounds gage the flow is constant; for final pressures higher than this the flow falls off at first slowly, and then more rapidly, till the final pressure equals the initial pressure, when the flow is, of course, zero. For final pressures higher than the critical pressure the flow is determined by a thermodynamic formula which need not be considered here, for we are concerned only with coil pressures lower than the critical pressure.

The temperature and pressure in the coils may be determined by again having recourse to the formula for heat transfer coefficient.

$$K = \frac{W(H_a - q_a)}{S \times T.\ D.}.$$

In the orifice method of operation the steam flow, W, is constant. The total heat in the steam "$H_a$" is constant. The

heat of the liquid of the condensed steam "$q_4$" increases only slightly with the coil pressure. Therefore, $(H_s - q_4)$, which is the heat removed by condensation in the coils, B.t.u. per pound of steam, is also practically constant. The surface, "S," is constant, and K, the coefficient of heat transmission, is variable, depending upon the condition of the coils as regards scale and certain other conditions of operation. The temperature difference may be solved for as follows:

$$T.D._c = \frac{W(H - q)}{K \times S}.$$

Since K is the only variable on the right side of this equation, it is evident that for every value of K there is a corresponding value of temperature difference required. The shell pressure fixes the shell temperature, and, with the temperature difference known, the coil temperature and coil pressure are easily found. The coil pressure will adjust itself to whatever value is required to produce a temperature difference which will be necessary to transmit the heat through the coils.

The operation of an evaporator with the orifice would therefore start with a high value of K and a very low coil pressure. As the operation continues and the value of K decreases the coil pressure will increase, and at the same time the steam flow, and therefore the capacity, remains constant.

### TEST OF DOUBLE EFFECT PLANT WITH ORIFICE.

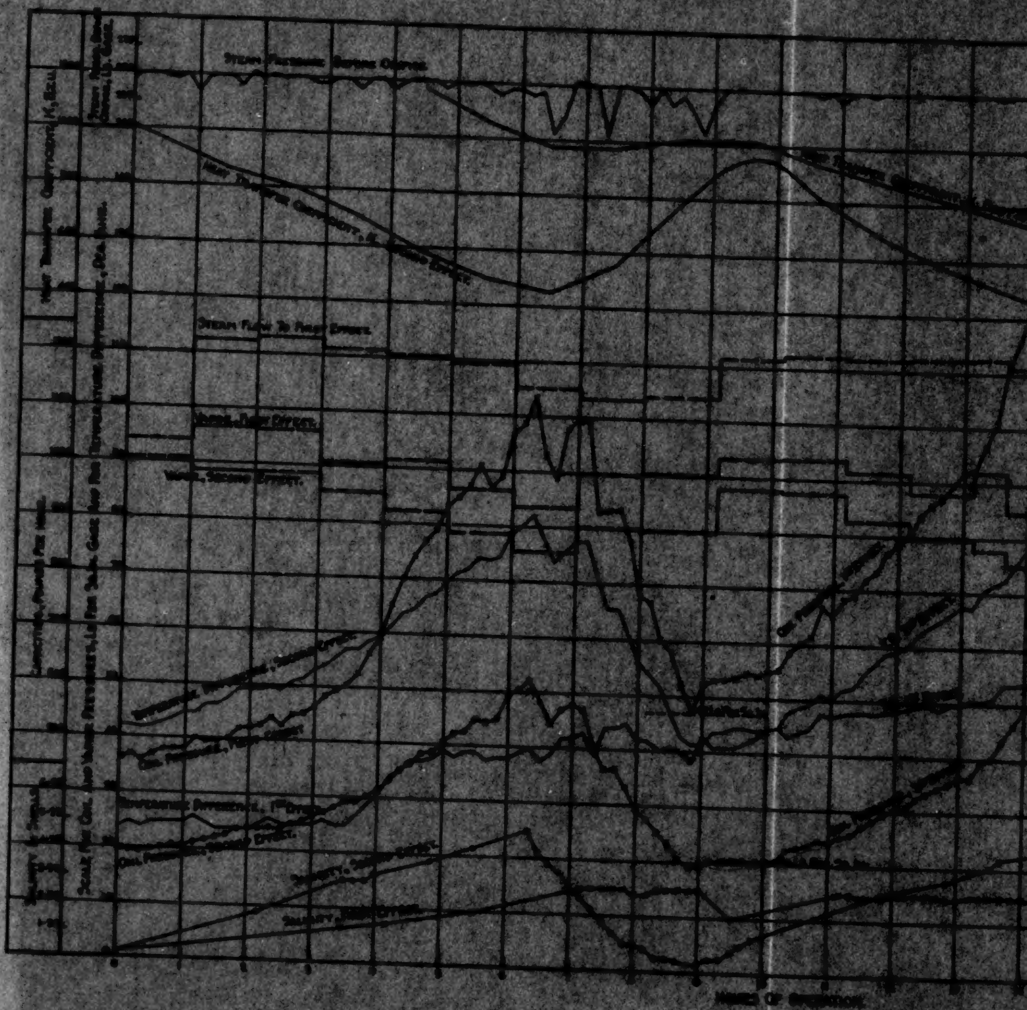
Having shown the theoretical action of the evaporator equipped with a steam orifice, the results of tests of the method will be given before discussing details of the application of the method. A preliminary trial was made upon a single-effect evaporator equipped with an orifice, and the action was exactly as was expected. At the start, when the coils were clean, the coil pressure was low, and as the operation continued the coil pressure slowly increased and the capacity remained constant. It was considered very desirable, however, to conduct the principal test upon a double-effect plant in view of

the wide use of the double-effect plant in the Naval Service and its added efficiency. A double-effect plant was therefore arranged for test. Each shell contained about 24 square feet of evaporating surface in the form of coils connected between manifolds as shown in the photographs, figures 7 and 9. A steam pressure of 250 pounds was available in the main line. The second-effect vapor discharged to a distiller condenser through a vapor feed heater, and atmospheric pressure was carried on the second effect shell. The vapor feed heater heated the feed for both effects to about 195 degrees, and the feed for the first effect was further heated by a coil drain heater. The capacity at which it was desired to operate was 120 gallons of water per day per square foot of total surface, which is double the present rating for Navy evaporators. From the expected efficiency of the double-effect plant this would require about 1,200 pounds of steam per hour to be supplied to the first effect coils. From Napier's formula, $W = \frac{PA}{70}$, for 250 pounds steam pressure in the line, the orifice required was computed to be 0.335 inch diameter. A brass plate 1/8 inch thick, with a hole in the center 0.335 inch in diameter, was placed between two flanges in the steam line to the coils of the first-effect evaporator.

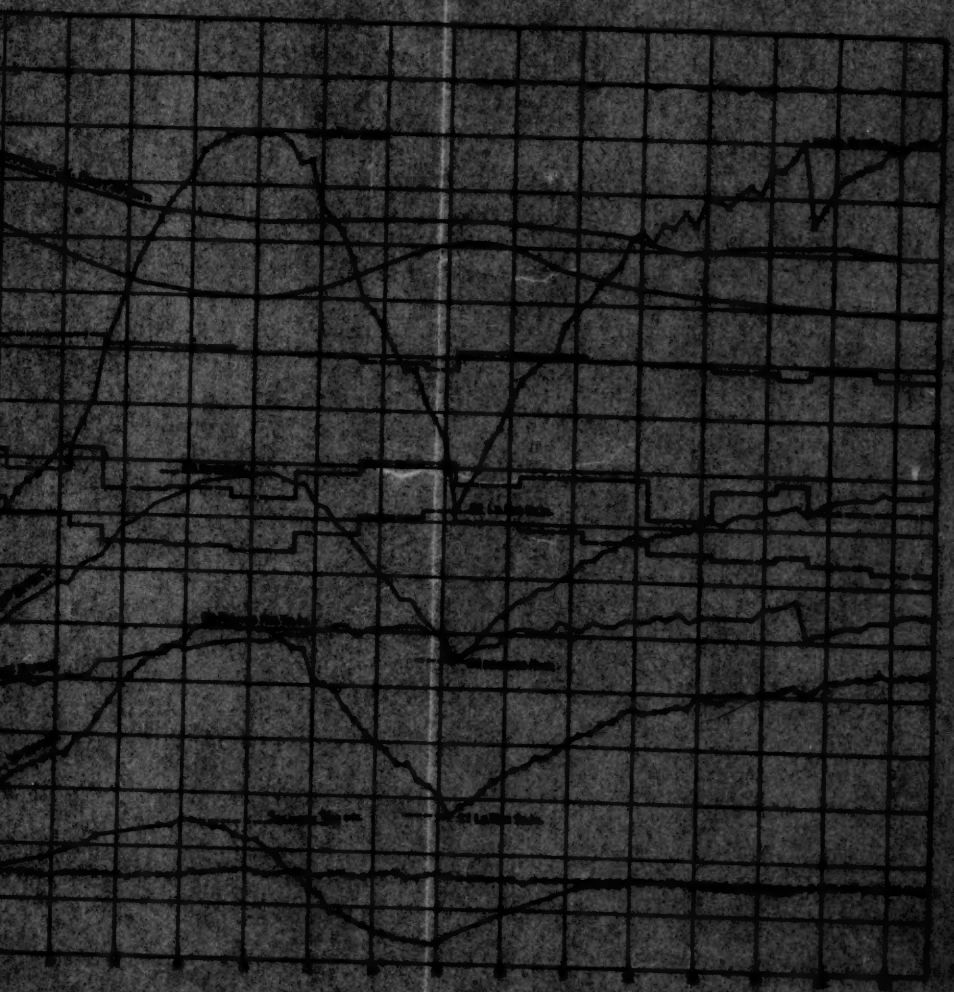
The full line pressure of 250 pounds was carried on the line leading to the orifice. Atmospheric pressure was carried on the second-effect shell. The water in each shell was carried at such a height as to submerge the coils. This level was determined by trial, as will be explained later. A water seal was kept on the coil drain by means of traps part of the time, and drain pots part of the time. This insured that all the steam entering the coils was condensed therein and only condensate allowed to discharge from the coils.

The log plot of the results of the first run is given in the insert, figure 5. On this are plotted the coil pressures, temperature differences, capacities, heat transfer coefficients and

salinities for each of the two effects. The salinity on the first effect was allowed to increase to 3/32nds and was maintained at that value throughout the test. The salinity of the second effect was varied, in order to observe the effect of a variable salinity on the action of the evaporator. The second-effect salinity was allowed to increase to 5/32nds, was then slowly decreased to that of Severn River water, which is less than 1/32nd, was increased to 5/32nds again, reduced to below 1/32nd, and increased to 3/32nds. Throughout the series of 26 hours the steam flow to the plant was constant except for a slight falling off when the line pressure fell below 250 pounds at the 6th hour. The pressure in the second-effect coil started at 20 pounds gage and increased to 52 pounds gage at the 6th hour, when the salinity was 5/32nds. Upon the reduction of the salinity the pressure fell and followed the salinity till at the end of the 26th hour the pressure was 55 pounds when the salinity was 3/32nds. The first-effect coil pressure started at 36 pounds gage and rose and fell, of necessity following the second-effect coil pressure (which was the first-effect shell pressure). A first-effect coil pressure of 100 pounds was reached at the 7th hour. It then decreased slowly to 47 pounds, increased to 150 pounds, decreased to 82 pounds, and reached 150 pounds at the 26th hour, with a salinity of 3/32nds on both effects. The temperature difference for the second effect varied, following the second-effect salinity, but the temperature difference for the first effect increased uniformly throughout the series from 24 degrees at the start to 65 degrees at the finish. *Throughout all this period the capacity of both effects was practically constant,* such variations as took place being due to variations in heat losses of the blow-down and the coil drain. (It should be noted that the run was not made non-stop. The plant was shut down each night and the salt water allowed to remain in the shell. In the morning the operation was resumed, and in each case the conditions obtained on starting in the morning corresponded



STEAM FLOW TO FIRST EFFECT
VAPOR, FIRST EFFECT
VAPOR, SECOND EFFECT
HOURS OF OPERATION



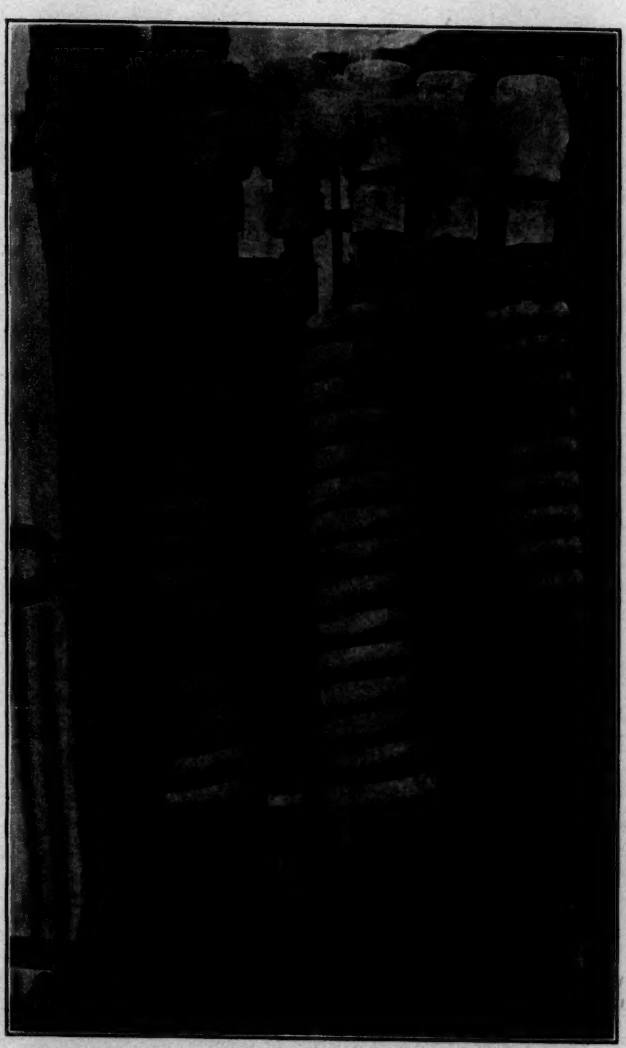
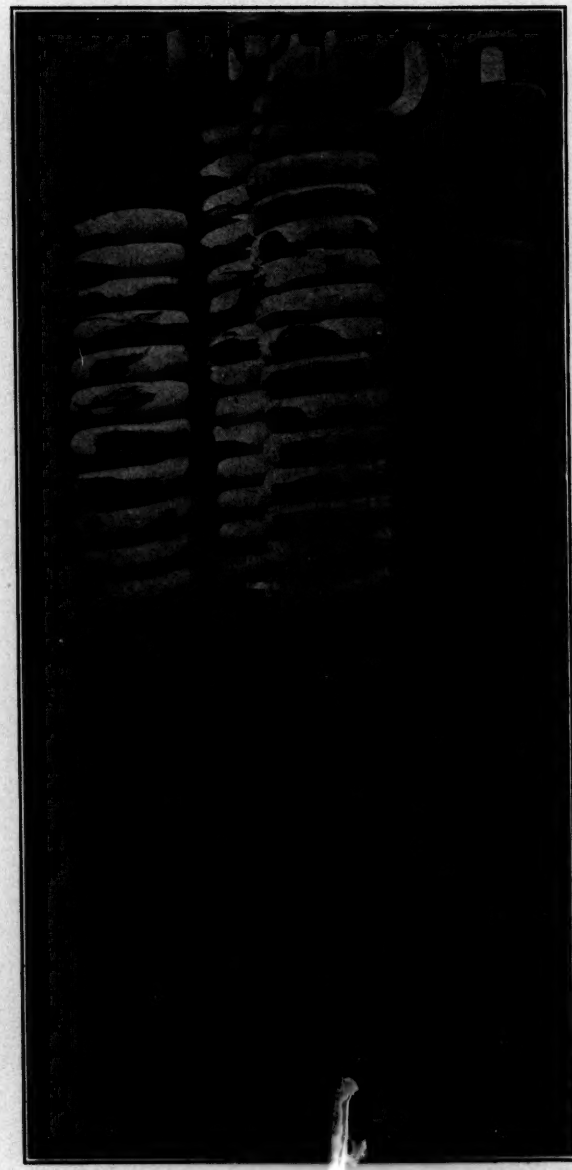


FIG. 7.—COILS, SHOWING SCALE AT END OF RUN WITH ORIFICE.



UPPER—FIG. 8.—COILS AFTER CRACKING SCALE, SECOND EFFECT.
LOWER—FIG. 9.—COILS AFTER CRACKING SCALE, FIRST EFFECT.

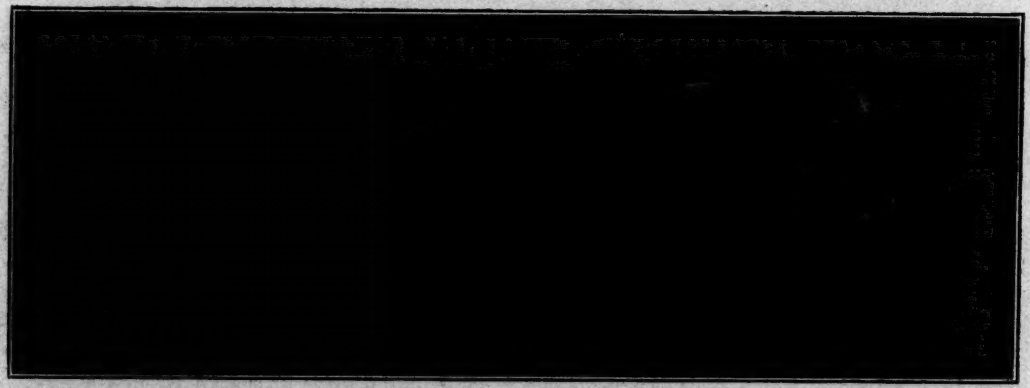


FIG. 10.—SCALE IN BOTTOM OF SHELL AFTER CRACKING SCALE.

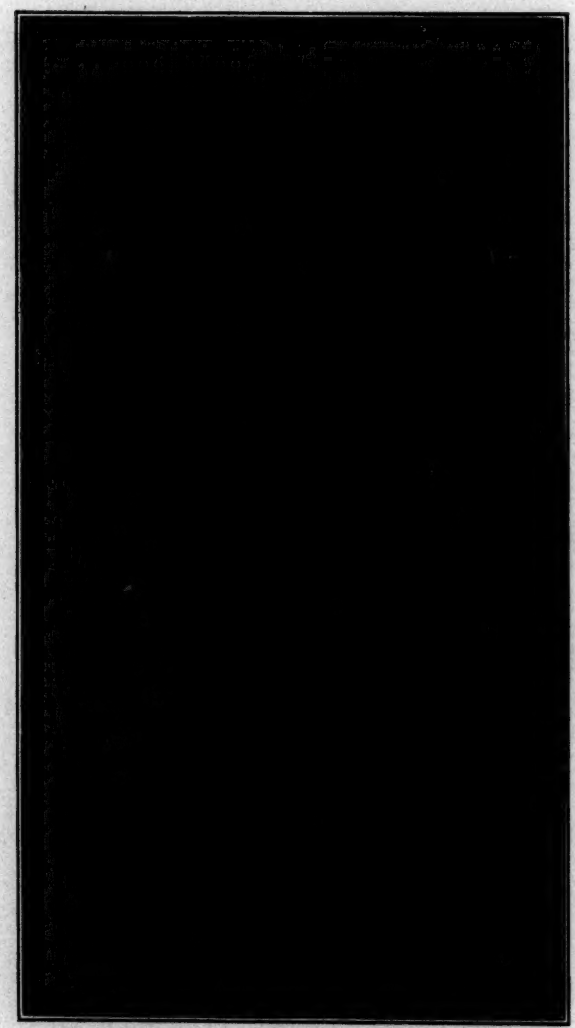


FIG. 11.—COILS, SHOWING SCALE PRODUCED BY LOW WATER LEVEL.

to the conditions which existed at the close of the previous day's run.) The average capacities obtained were as follows:

| | |
|---|---|
| Steam used | 1,196.6 lbs. per hr. |
| Vapor produced, 1st effect | 984.6 lbs. per hr. |
| Vapor produced, 2nd effect | 896.4 lbs. per hr. |
| Total vapor | 1,881.0 lbs. per hr. |
| Total vapor | 543. gals. per day |
| Total vapor | 112. gals. per day per sq. ft. of surface |
| Total vapor | 1.575 lbs. per lb. of steam |

The capacity produced was almost double the present rating, and there was no variation in capacity with variations of salinity from less than 1/32nd to 5/32nds and variations of first-effect coil pressure from 36 pounds to 150 pounds per square inch. No regulation of coil pressures was required. The vapor produced was of a purity of less than 1/2 grain chlorine per gallon at all times.

When the coil pressures reached the maximum values, operation was ceased for the removal of scale. Before removing scale, however, the second effect evaporator was opened and the photograph shown in figure 7 was taken, which shows a thin, even coating of scale on the coils. The evaporator was immediately closed and the scale was removed by a scale-cracking process which consisted of surrounding the coils with cold water and admitting 40 or 50 pounds steam pressure suddenly into the coils, and at the same time keeping the coils cleared of condensate and air by watching the drain pot. Upon opening the evaporator it was found that a large part of the scale had been cracked from the coils and was seen lying in the bottom of the shell. Fragments gathered from the shell are shown in figure 10.

The result of the removal of scale is an increase in the heat transfer coefficient, and upon resuming operation after removing scale the coil pressures are low and the cycle may be repeated. The question naturally arises as to the time required

for the pressures to increase to such values as to necessitate the removal of scale, and the influence of rate of evaporation and other operating factors upon this rate of increase of pressure. The temperature difference in the first effect increased at an average rate of 1½ degrees per hour. The coil pressure and heat-transfer coefficient may be determined from the increase in temperature difference.

The effect of salinity on temperature difference and heat transfer, as determined from the second-effect data, is given on figure 6. The temperature difference is increased about 9 degrees per 32nd increase in salinity and the heat-transfer coefficient is correspondingly reduced. At 1/32nd salinity the value of K is 1,200, at 3/32nd it is 780 and at 5/32nd it is reduced to 570. These values apply for clean coils, the results having been corrected for the increase in temperature difference due to scale formation. When operating by the constant-pressure method these changes in salinity would have an enormous effect upon the capacity, but in the orifice method the temperatures and pressures are automatically adjusted to maintain the constant capacity regardless of salinity.

A second series of runs was made at the same capacity as the first series, but with a constant salinity of 3/32nd in both effects, using continuous blow-down. During this run an average total capacity of 108 gallons per day per square foot of coil surface was obtained for a period of 30 hours, during which time the first-effect coil pressure increased slowly from 44.0 pounds per square inch gage to 160.0 pounds, and the second-effect coil pressure increased from 20.4 pounds to 75.0 pounds. The average rate of increase in temperature difference for the first effect was 1 degree per hour and for the second effect was 1.6 degrees per hour. At the end of this period the scale on both effects was cracked by suddenly admitting steam to the coils which were surrounded by cold water. Operation was immediately resumed, after filling the shell with brine of 3/32nd salinity without opening the shell,

and the coil pressures were found to have been reduced to 88.1 pounds on the first effect and 42.2 pounds on the second effect. The pressures then gradually increased at the same rate as previously.

A third series of runs was made with a smaller orifice installed, in order to determine the action at a lower capacity. This orifice was 0.258 inch in diameter and discharged 680 pounds of steam per hour with 250 pounds line pressure. The start was made with clean coils which produced a pressure of 25 pounds on the first-effect coils and 15 pounds on the second-effect coils. Operation continued for a period of 91 hours during which the coil pressure slowly increased to 165 pounds on the first effect and 55 pounds on the second effect. The temperature difference on the first effect increased at a rate of 0.6 degree per hour and on the second effect the increase was 0.5 degree per hour. In each series it was noticed that the rate of increase in temperature difference was smaller during the latter part of the run when scale had formed on the coils and the coil pressures were relatively high, than during the first part of the run when the coil pressures were lower. This disproves a prevalent idea that high coil pressure should not be used on account of unfavorable effects of high coil pressures upon scale formation.

The capacity obtained averaged 64.5 gallons per day per square foot of surface, which is above the present rating of evaporators. The capacity was slightly higher at the start than at the end of the run on account of changes in efficiency due to coil drain losses.

The scale was cracked by the sudden application of steam when the coils were surrounded by cold water. The appearance of the second-effect coils after cracking scale is shown by the photograph, figure 8. It was feared that possibly the scale cracking would not be successful on account of the larger accumulation of scale as indicated by the low heat-transfer coefficient. On the second-effect coils a large portion of the scale

was entirely removed, and most of the scale remaining was in a very loose condition and could be removed with the finger tips or with very little effort. Some of the scale, though loosened, is held in position on account of the close winding of the coils. The coil at the extreme right, which is not wound so closely as the others, shows a better removal of scale. The first-effect coils were also satisfactorily scaled, as shown in figure 9. There is no doubt that this method of removing scale from this type of coils can be developed on shipboard so that the scale may be removed at the end of each period and the operation continued without the necessity of opening the evaporator door, except at long intervals for inspection. A small door should be provided in the lower part of the shell for the purpose of hauling out the scale which falls off. This method of removing scale is, however, not an essential part of the method of operating with an orifice, for the scale may be removed by hand, if desired, when it has reached a condition which demands it.

The orifice method of operation provides a convenient method of investigating the effect of water level, and also of determining the proper level at which to operate. It must be recognized that the level in the glass does not represent the brine level in the shell on account of the boiling and foaming of the brine in the shell, the level in the shell being always considerably higher than the level as indicated by the glass. In fact, the level in the shell has probably a very indefinite value, varying considerably with vapor pressure, feed temperature, salinity and capacity. However, the level in the glass may be used as an indication of the conditions in the shell. A series of runs was made with a constant salinity of 3/32nds in which the water level was varied from 24 inches to 8 inches in the glass, measured above the center of the lower manifold. A run was made for one hour at each level, by 2-inch increments, and the results are plotted in figure 12. As the level was decreased, the coil pressure, and therefore the temperature

difference, slowly increased, till a level of 12 inches was reached, when the pressure went up much more rapidly. The higher temperature differences indicate that a part of the surface is exposed, because the amount of heat transmitted is the same at all levels. In addition to the higher coil pressures required, there is another very decided objection to a low water level. That is the deposit of a large amount of scale on the upper part of the coils. Figure 11 shows the appearance of the coils after operating only 8 hours at a level of 8 inches. The scale is so heavy as to fill in the spaces between the convolutions of the coils. Contrasting this with figure 7, which shows the nature of the scale formed with a high water level, the advantage of high water level is clearly seen. The excessive formation of scale at low water levels is easily accounted for. When the level is low the boiling causes a large amount of brine to be splashed up against the exposed portion of the coils, and as the brine hits the hot coils the water is evaporated and all the salt which was in the water is deposited on the coils, leaving a heavy scale. When the water level is carried high enough to keep the coils surrounded with water a large proportion of the salt stays in solution and is blown out of the shell with the blow-down.

If, after starting with a low level, the level is raised, the excessive scale formed on the upper parts of the coils reduces the heat-transfer coefficient to such an extent that either the desired capacity cannot be obtained or the coil pressure must be increased. With the usual method of operation with constant coil pressure, on starting with clean coils it is generally necessary to start with a low level in order to prevent priming which would result from the high capacity produced at a high level. A heavy scale rapidly forms over a large portion of the coils, thereby preventing the proper transmission of heat, even should the water level be later raised. One fundamental advantage of the orifice method may be seen in the ability to carry a high water level at all times without the danger of

priming, and the consequent production of thin, uniform scale, which may be easily removed, rather than the production of heavy scale similar to that shown in figure 11.

The only objection to a high water level is the danger of carrying over salt. At all levels up to 24 inches the vapor was free of chlorine, and in any properly designed evaporator the level can be carried so as to cover the coils without priming. The high level keeps all of the heating surface in action, reduces the amount of scale, requires lower pressures and increases the time of running without cracking scale. Also, on account of the larger volume of water in the shell, the blow-down period is increased if intermittent blow-down is used. The conclusion is that the level should be carried so as to submerge all the surface, if this is possible, without making chlorine in the vapor. The level at which to operate any evaporator equipped with an orifice may be quickly determined by a short series of runs similar to that plotted in figure 12. In determining the proper level in any specific case it will be well to first try operation with a level in the glass which is 4 or 5 inches below the top of the evaporating surface. If chlorine is produced in the vapor with this level the level must be lowered. The coil pressure should be observed as the level is lowered, and it should be possible to establish a working level at a point which does not produce a greatly increased coil pressure. In most evaporators there will be a considerable range in level over which the coil pressure is nearly constant and the vapor is sufficiently free from chlorine. The working level should be established midway in this range of levels.

### COMPARISON OF ORIFICE METHOD WITH USUAL METHOD.

Figure 13 has been prepared to show the comparison between the action of the usual constant coil-pressure method and the improved-orifice method of operation for a single-effect plant. The shell pressure is assumed to be atmospheric and the line pressure is 150 pounds gage. The left-hand ordinates are steam flow through orifice or steam used, pounds per

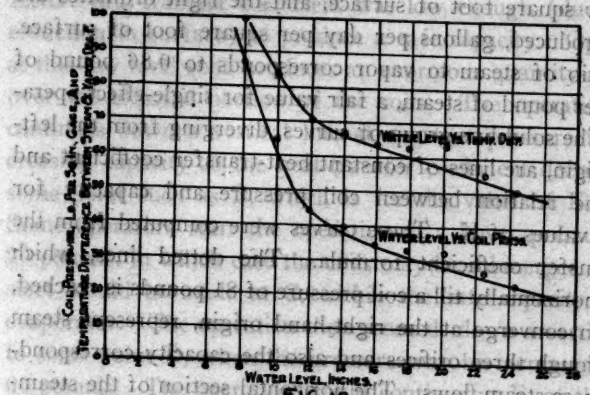

Water Level Vs. Tank Dial
Water Level Vs. Coil Pressure
Water Level, Inches.

FIG 12.

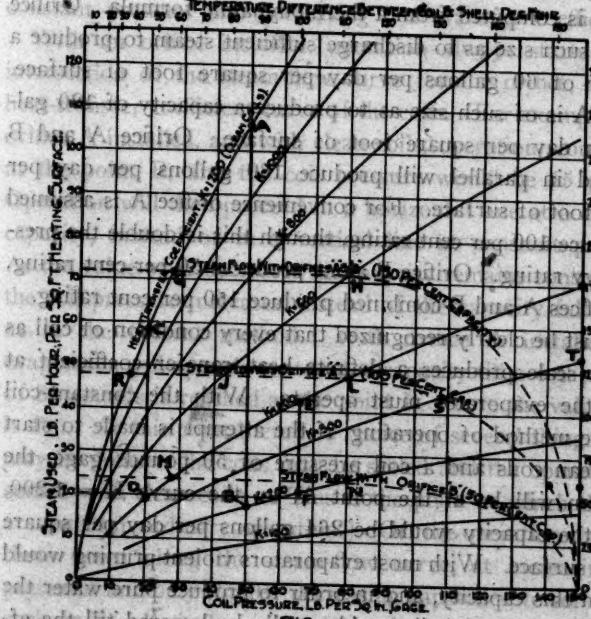

Temperature Difference Between Coil & Shell, Deg. Fahr.
Steam Used, Lb. Per Hour, Per Sq. Ft. Heating Surface.
Vapor Produced, Gal. Per Day, Per Sq. Ft. Surface.
Coil Pressure, Lb. Per Sq. In. Gage.

FIG 13.

hour per square foot of surface, and the right ordinates are vapor produced, gallons per day per square foot of surface. This ratio of steam to vapor corresponds to 0.86 pound of vapor per pound of steam, a fair value for single-effect operation. The solid-line group of curves, diverging from the left-hand origin, are lines of constant heat-transfer coefficient and show the relation between coil pressure and capacity for various values of K. These curves were computed from the heat-transfer coefficient formula. The dotted lines, which extend horizontally till a coil pressure of 81 pounds is reached, and then converge at the right-hand origin, represent steam flow through three orifices and also the capacity corresponding to these steam flows. The horizontal section of the steam-flow curves are computed from Napier's equation, and the curved portion, from the critical pressure of 81 pounds to 150 pounds, is computed from a thermodynamic formula. Orifice B is of such size as to discharge sufficient steam to produce a capacity of 60 gallons per day per square foot of surface. Orifice A is of such size as to produce a capacity of 120 gallons per day per square foot of surface. Orifice A and B arranged in parallel will produce 180 gallons per day per square foot of surface. For convenience orifice A is assumed to produce 100 per cent rating, though this is double the present Navy rating. Orifice B alone produces 50 per cent rating, and orifices A and B combined produce 150 per cent rating.

It must be clearly recognized that every condition of coil as regards scale produces a definite heat-transfer coefficient at which the evaporator must operate. With the constant-coil pressure method of operating, if the attempt is made to start with clean coils and a coil pressure of 50 pounds gage, the operation will be at the point A, on the curve K = 1,200, where the capacity would be 264 gallons per day per square foot of surface. With most evaporators violent priming would occur at this capacity, and in order to produce pure water the water level in the shell would usually be lowered till the ef-

fective heat-transfer coefficient was low enough (due to the reduction in surface) to produce a safe capacity. Operation at this low level soon produces an enormous quantity of scale over the exposed portion of the coils. The operation then continues down a vertical line at constant pressure and decreasing capacity as the heat-transfer coefficient decreases to some point such as B, which is for the value of K = 200. Some operators start the operation at a lower pressure, say 20 pounds, in which case the coils may be submerged and the start would be made at C, within the safe capacity of the evaporator. The operation then proceeds down a vertical line to some point D, when the pressure may be increased to say 66 pounds. The operation would continue at E, at the same value of K as existed for the lower pressure, and from E the capacity would again decrease as the operation proceeded at constant pressure and decreasing heat-transfer coefficient.

With an orifice installed, operation always starts at the intersection of the steam-flow curve for the orifice, and the heat-transfer coefficient curve of the coils. For example, with orifice A installed, and with clean coils, operation would start at the point R, and the coil pressure would be 15 pounds. Operation will then proceed in a horizontal line at a constant capacity, instead of in a vertical line at constant pressure. As the value of K decreases the capacity remains constant and the coil pressure increases, instead of the capacity falling off, as in the present method. With orifice A installed, the evaporator will operate at its rated capacity till a coil pressure of 81 pounds is reached at the point L, where the value of K is 400. At this point there is a choice of several things which may be done. The coils may be scaled and the operation resumed with clean coils. (This is the point at which the coils are scaled in ordinary operation.) Or, the operation may continue down along the steam-flow curve, at increasing coil pressures and only slightly decreasing capacities, to some point such as S, where the coil pressure is 110 pounds, K is 300,

and the capacity is reduced only 10 per cent below the rated capacity. If there is no objection for structural reasons to carrying full-line pressure in the coils and steam header, the by-pass valve may be opened wide, allowing the evaporator to operate with full-line pressure of 150 pounds on the coils. This will cause the operation to move up the line of K = 300, bringing the operation to the point T, at which the capacity is only 10 per cent above the rated capacity and where the operation will therefore be safe and without danger of priming. From T, the capacity will fall at constant coil pressure as the scale further accumulates. If the coil pressure may not be increased above 81 pounds, and it is desired to continue running when this pressure is reached, orifice A may be closed and orifice B opened. This will mean dropping back along the line of K = 400 to the point M, where the coil pressure is only 27 pounds, and 50 per cent capacity is obtained. Operation may now continue at 50 per cent capacity along the line M N till the value of K is reduced to 200 at a coil pressure of 81 pounds.

When it is desired to force the evaporator orifices A and B may be operated in parallel at the start. This will result in starting at the point G with 150 per cent of rating and continuing to the point H where K = 600. From this point the operation may be either continued along the steam-flow curve at increasing coil pressures and slightly decreasing capacities, or the operation may be immediately resumed at the rated capacity by closing off orifice B, and dropping back to the point J and continuing at rated capacity.

### ARRANGEMENT OF ORIFICE INSTALLATION.

The proposed arrangement of the steam piping to the first-effect coils for orifice operation is shown by the sketch, figure 14. The three-branch construction of standard fittings is substituted for the reducing valve at the coil entrance. The upper branch contains the orifice plate "A" for usual operation at

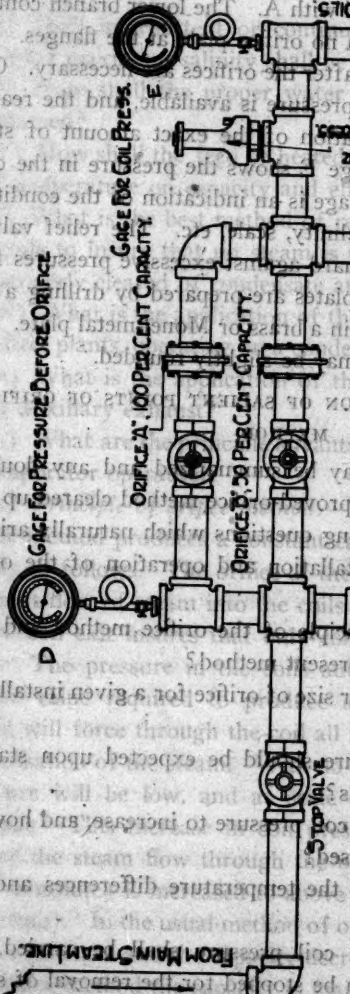



FIG. 14.—SKETCH OF ORIFICE INSTALLATION.

100 per cent capacity. The center branch contains the orifice plate "B" for operation at 50 per cent capacity, or at 150 per cent when used in parallel with A. The lower branch contains the by-pass valve C, with no orifice plate at the flanges. The gages shown before and after the orifices are necessary. Gage D shows if the full-line pressure is available, and the reading of this gage is an indication of the exact amount of steam flowing to the coils. Gage E shows the pressure in the coils, and the reading of this gage is an indication of the conditions in the shell as regards salinity, scale, etc. The relief valve is necessary, as usual, to guard against excessive pressures from any cause. The orifice plates are prepared by drilling a hole of the required diameter in a brass or Monel-metal plate. The inlet edge of the orifice may be slightly rounded.

## SUMMARIZED DISCUSSION OF SALIENT POINTS OF ORIFICE METHOD.

The entire subject may be summarized and any doubtful points concerning the improved-orifice method cleared up by a discussion of the following questions which naturally arise in connection with the installation and operation of the orifice method:

(*a*) What is the principle of the orifice method and how does it differ from the present method?

(*b*) How is the proper size of orifice for a given installation determined?

(*c*) What coil pressure should be expected upon starting operation with clean coils?

(*d*) What causes the coil pressure to increase, and how are the coil pressures decreased?

(*e*) At what rate do the temperature differences and coil pressures increase?

(*f*) What maximum coil pressure shall be carried, and when shall the operation be stopped for the removal of scale?

(*g*) What method shall be used in removing scale so that the operation may be continued without opening up the shell?

(*h*) What vapor pressure shall be carried on the lowest-effect shell?

(*i*) Shall intermittent or continuous blow-down be used, and what maximum salinity shall be carried in the shell?

(*j*) How shall the proper water level be determined and maintained?

(*k*) How shall the feed be heated, and what is the effect of feed temperature on capacity and efficiency?

(*l*) What is the best method of removing condensate from the coils to insure that no steam is wasted and that the coils are properly cleared of condensate and air?

(*m*) What is the application of the orifice method to multiple-effect plants, operating on a condenser and with a vacuum?

(*n*) What is the application of the method to evaporators using auxiliary exhaust?

(*o*) What are the principle advantages of the orifice method of evaporator operation?

(*a*) *Principle of orifice method.*—The manner in which the orifice method produces a constant capacity may be explained briefly as follows. The orifice at the coil entrance produces a constant flow of steam into the coils. The water seal or trap at the coil exit insures that only condensed steam leaves the coils. The pressure in the coils automatically adjusts itself to the value required to produce a temperature difference which will force through the coil all the heat required for the condensation of the steam. With no scale on the coils the coil pressure will be low, and as scale forms the pressure will increase. This increase in coil pressure, however, does not reduce the steam flow through the orifice until the coil pressure (absolute) is increased to above 0.58 of the line pressure (absolute). In the usual method of operation the coil pressure is held constant and the capacity decreases as scale forms. In the orifice method the capacity is maintained constant and the coil pressure increases as scale forms.

(*b*) *Size of orifice required.*—The size of the orifice required is determined from the capacity desired and the pressure

of steam available. The capacity may be taken as the maximum capacity which the evaporator will produce without priming, or it may be based upon a certain capacity desired per square foot of surface. The steam required is computed from the known steam economy of the plant, pounds of steam required per pound of vapor produced, or from the expected economy, considering number of effects, feed temperature, blow-down losses, etc. For single-effect plants the steam economy will vary from 1.20 to 1.35 pounds of steam per pound of vapor, and for double-effect plants the steam economy will vary from 0.60 to 0.75 pound of steam per pound of vapor. If available, the high-pressure steam line should be used. The size of orifice for producing 100 per cent capacity is then computed from Napier's formula,

$$W = \frac{PA}{70},$$

in which,

W = steam required, pounds per second.

P = steam pressure, pounds per square inch, absolute, before orifice.

A = area of orifice, square inches.

It is recommended that an additional orifice of 50 per cent capacity be installed and that the two orifices and a by-pass for the purpose of cracking scale, be installed in parallel as shown in the sketch, figure 14. No reducing valve is required, because in the orifice method, the coil pressures are automatically regulated. The gages are necessary for the intelligent observation of the performance. The gage at the orifice entrance measures the steam flow and the gage at the coil entrance indicates the conditions in the shell as regards scale, etc. The relief valve is required, as usual, which should be set for the highest allowable working pressure of the coils. If the orifice installed does not produce the capacity desired, on account of some condition which was unknown when computing the size of the orifice, the diameter of the orifice may

be easily increased by reaming till the desired capacity is obtained. The thickness of the plate used should be about 1/8 or 3/16 inch.

(*c*) *Initial coil pressure.*—The coil pressure which should be expected upon starting with clean coils may be determined from a consideration of the heat-transfer coefficient formula,

$$K=\frac{(H_s-q_d)W}{S\times T.\ D.},\ \text{or transposing,}\quad T.D.=\frac{(H_s-q_d)W}{S\times K}.$$

The value of $(H_s-q_d)$, heat removed by the condensation of steam in the coils, B.t.u. per pound, is nearly 1,000. The value of the heat-transfer coefficient K, is also approximately 1,000 for clean coils and 3/32nds salinity. Substituting,

$$T.D.=\frac{W\times 1{,}000}{S\times 1{,}000},\ \text{or}\ T.D.=\frac{W}{S}.$$

The temperature difference to be expected on starting with clean coils is therefore approximately equal to the pounds of steam per hour per square foot of surface. The coil pressure corresponding to the temperature difference is determined from the principle that the pressures in the shell and coils correspond to the temperatures.

(*d*) *Increase in coil pressure.*—The main factor causing the gradual increase in coil pressure is the gradual accumulation of scale on the coils. The coil pressure is also increased by increase in vapor pressure, increase in salinity of brine, low water level, and air binding in the coils. When the coil pressure has increased to its maximum value, attention should first be given to the drain pot to see that the coils are properly drained of air and condensate. Then it must be ascertained whether the water level is at the proper height. The salinity should be taken, and if above 3/32nds it should be reduced by blowing down. If the coil pressure remains high after giving proper attention to these points the scale must be removed from the coils either by hand or by the scaling method described herein.

(*e*) *Rate of increase in temperatures and pressures.*—The tests showed an increase of temperature difference of 1 to 1½ degrees per hour at a capacity of 110 gallons per day per square foot of surface. At a capacity of 64.5 gallons per day per square foot of surface the rate of increase was ½ degree per hour. In every run it was noticed that the rate of increase was smaller toward the end of the run, when the pressure was high, than at the start. In operating with the orifice method a careful log should be kept of the increase in pressure and temperature, and every effort should be made to adopt methods of operation which will keep this rate of increase of temperature difference as small as possible, as the increase in temperature determines definitely the length of time a desired capacity may be maintained. The rate of increase of temperature difference is influenced by probably many conditions of design and operation, and further developments in the orifice method will be along lines of obtaining the smallest possible rates of increase of pressures and temperature differences.

(*f*) *Maximum coil pressures.*—The first-effect coil pressure (absolute) may be allowed to increase to 0.58 of the line pressure without any decrease in steam flow. The maximum coil pressure is limited only by the safe working pressure of the coils and steam manifolds. For these reasons it is advisable to use the highest line pressure available for the initial pressure at the orifice entrance. If 150 pounds line pressure is used, the coil pressure in the highest effect may increase to 81 pounds without decrease in steam flow, but with 250 pounds line pressure, the coil pressure may be allowed to increase to 140 pounds (if this is a safe working pressure of the coils) without decrease in steam flow. The use of the highest available pressure simply means that the operation at any given capacity may be continued to lower values of heat-transfer coefficient, and therefore for a longer period without cracking scale. With the usual method, high coil pressures are not

advised on account of the danger of priming, but with the orifice method, where the capacity is fixed, this danger does not exist, and there appears to be no objection to allowing the operation to continue until the coil pressure has increased to the allowable working pressure of the coils. In double-effect plants the second-effect coil pressure must not be allowed to increase above the maximum allowable first-effect shell pressure.

From figure 13 it is seen that the heat-transfer coefficient which exists when the operation must be stopped on account of reaching a maximum pressure depends upon the capacity at which the evaporator is being operated. Assuming a maximum allowable coil pressure of 80 pounds gage, at 150 per cent rating the limit of operation occurs when the heat-transfer coefficient is 600. At 100 per cent rating, the limit of operation is at a value of K = 400 and at 50 per cent of rating (60 gallons per day per square foot of surface) the operation need not cease till the value of K has been reduced to 200. Thus, at the lower rates of evaporation the operation may be continued much longer.

(*g*) *Removal of scale.*—When the coil pressures have increased to the maximum permissible values at any desired rate of evaporation the scale must be removed from the coils either by hand or by a self-scaling method. The orifice method of operation permits of the carrying of a high water level, and this results in the production of a uniform and relatively thin coating of scale over the entire surface of the coils. It has been repeatedly demonstrated that this sort of scale may be effectively removed from coils by the self-scaling method described herein. The method consists of covering the coils with cold water and suddenly admitting a steam pressure of 50 or 60 pounds on the coils through a by-pass steam connection. The coils must be kept thoroughly clear of condensate and air during this operation by watching the drain pot carefully. This is necessary in order to bring up the tem-

perature of the metal of the coils as rapidly as possible. As soon as the water in the shell becomes warm the steam should be turned off the coils, to prevent the evaporation of the water at a high rate and the carrying over of salt. In some cases it may be necesary to repeat the operation to effect a thorough cleaning. The principle of the method is the difference in expansion of the metal of the coils and the scale. Although this method of cracking scale has been demonstrated as effective only on coils of the types shown in the photographs, it may possibly be successfully applied to other types of evaporators with a little development.

The coil pressure obtained upon resuming operation after cracking scale is an indication of the thoroughness with which the scale has been removed. A sludge door should be provided for hauling out the scale which is cracked off. This door is also convenient for washing off the coils with a hose and for inspection of coils.

(*h*) *Shell Pressures.*—At present it is sometimes considered necessary to hold a pressure of 5 pounds on the lowest-effect shell to prevent priming. This has probably been necessary to reduce the temperature difference when carrying a constant coil pressure. When using the orifice method the temperature differences adjust themselves, and all of the tests made with the orifice method prove there is no reason for carrying higher than atmospheric pressure in the shell of the lower effect. The valve between the lowest-effect shell and the distiller should be opened wide, and with atmosphere pressure on the distiller, the shell pressure may be slightly above the atmosphere. In a two-effect plant, the first-effect shell pressure will increase slowly and the operation must be stopped when a pressure is reached which is limited by the highest safe working pressure fixed by the strength of the shell. Relief valves should be fitted as usual to prevent exceeding the maximum safe pressure.

(i) *Blow-down.*—With sea water as feed the blow-down period may be computed from the formula,

$$P = \frac{V(S-1)}{W},$$

in which,

P = blow-down period, hours.

V = pounds of water in the shell at the operating level.

S = salinity at end of blow-down period, pounds of salt per 32 pounds of water (32nds).

W = vapor produced, pounds per hour.

A maximum salinity of 3/32nds is usually recommended. Certainly the blow-down should never take place at lower salinities than this, on account of the unavoidable loss of heat in the brine blown out. In the orifice method of operation the salinity may, if desired, be carried slightly higher than 3/32nds before blowing down, for the following reason. With the orifice method a high salinity merely increases the coil pressures, whereas in the operation with constant coil pressure, the increase in salinity reduces the capacity on account of the decrease in heat-transfer coefficient. If the capacity and volume of shell are such as to require too frequent a blow-down, as computed from the above formula, intermittent blow-down cannot be recommended. The blow-down should then be continuous, and the salinity regulated by the use of a salinometer. A semi-continuous blow-down consisting of blowing down a certain amount of brine, amounting to 5 or 6 inches in the glass, at regular intervals, may be used as an alternative. There are many instances where evaporators are operated without the use of salinometers, and the salinities are allowed to increase to beyond the saturation point of the brine, with disastrous effects upon scaling of coils, capacity, and purity of vapor. It is absolutely impossible to obtain satisfactory evaporator operation without giving proper attention to this important feature. With the orifice method, on

account of the constant capacity feature, it is possible to compute accurately the proper blow-down period. If intermittent blow-down is possible, it is convenient to combine with the blow-down a partial scale cracking. After the brine has been blown out the shell may be filled to the working level or slightly higher with cold water (instead of the heated feed), and a steam pressure of 50 pounds put on quickly through the by-pass. In a double-effect plant a live steam connection should be made to the second-effect coils for this purpose. As soon as the water in the shell becomes warm the by-pass should be closed, and operation continued with the orifice. It will be found that this will produce at least a partial cracking of the scale.

(*j*) *Water Level.*—In the present constant-coil pressure method of operation it is frequently impossible to commence the operation with a water level sufficiently high to cover the coils. A low level must be carried to avoid priming, with the result that the upper portion of the coils are quickly covered with a heavy scale, the capacity is decreased, and the removal of scale is made difficult. In the orifice method the proper level may be easily determined and carried at all times. In most types of evaporators a level 4 or 5 inches in the glass below the top of the evaporating surface will cover the surface with brine and be safe against priming. A definite rule for the water level cannot be given, because the proper level will be different in different designs of evaporators and also at different capacities and vapor pressures in the same evaporator. A large capacity, on account of increased bubbling and foaming in the shell, will possibly require a lower level in the glass than smaller capacities, while a higher vapor pressure will allow a higher level to be carried. The point is that with the orifice method a certain proper level for a given capacity may be easily determined, and this level will be such as to cover the coils, and thereby reduce enormously the formation of scale.

(*k*) *Feed Heating.*—In the orifice method the steam flow is constant, but the efficiency, and therefore the quantity of vapor produced, depends upon the temperature of the feed. Inasmuch as the vapor produced from the last effect must be condensed, there appears to be no excuse for not using a part of this vapor in heating the feed to as high a temperature as possible. The best arrangement for heating the feed is unquestionably that recommended by Commander Dinger,* which consists of: "first allowing the feed to pass through the vapor heater, where it may be heated up to the temperature of the vapor, about 210 degrees F., and then to cause the feed to the first-effect shells to pass through a first-effect coil drain heater where it can be heated to the temperature of about 270 degrees F. In a double-effect plant, with this arrangement, the feed water can enter the evaporator shells at about 210 degrees for the second effect and 270 degrees for the first effect, where the initial pressure in the H.P. coils is 50 pounds." It would be very desirable to have the vapor heater arranged in series with the distiller condenser, so that all the vapor from the second effect passes through the heater. Adequate heating of feed saves not only in reducing the heat required for producing vapor, but reduces enormously the blow-down losses.

(*l*) *Removal of condensate.*—Either a trap or a drain pot may be used to remove the condensate from the coils. In either case it is absolutely essential to have a water glass which records the level of the water seal and which is in plain sight at all times to insure that no steam is blowing through. A water glass on the steam manifold or steam header of the evaporator is sometimes used, but this is wholly unsatisfactory. The condensate must be drawn off into a separate vessel, consisting of either a trap or drain pot, to insure that the coils are free from condensate. If a drain heater is used it should be located beyond the drain pot. If a heater should be included as part of the drain pot, live steam would be drawn from the

* "Marine Feed Water Heating," A. S. N. E. JOURNAL Feb., 1914, P. 152.

coils to heat the feed and therefore no advantage would be gained. A pressure must be maintained on the drain heater in order to heat the feed above 212 degrees. This pressure is most easily obtained by regulating the drain-pot water level from a point beyond the heater. *The air vent on the drain pot should be provided and used.* It is just as necessary to remove the air which collects in the coils as a result of the condensation of the steam as it is to remove the condensate. If the coils become filled with air, or air bind, the heat cannot be transmitted through the coils and excessive coil pressures will result.

In designing the drain pot it should be noted that the larger the horizontal sectional area of the drain pot the easier will be the control of the level in the glass. A number of drain pots have been seen whose unsatisfactory action has been due to their relatively small sectional area.

(*m*) *Multiple effect application.*—Three or four-effect operation with the lowest effect operated at a high vacuum is at present very troublesome because of the difficulty of preventing priming in the lower effects. This is largely due to the following condition. At the lower pressure the temperature difference due to a small pressure difference is comparatively large, and it is almost impossible to know at what pressures to operate to produce a given capacity without the danger of priming. Also, a small change in vacuum produces a considerable change in temperature difference which induces priming. At the high vacuua at which the lower effects operate the specific volume of the steam is very high, and on account of the consequent high velocities of vapor at the disengaging surface it is to be expected that priming will occur at relatively low capacities, unless the disengaging surface and baffling have been especially designed for the high vacuum operation.

Having established the maximum capacity which the lower effects may produce without priming, the orifice which sup-

plies steam to the first-effect coil may be designed for a steam flow which will produce this capacity. The air must be removed from the coils which are under a vacuum by leading a small connection from the top of the drain pot or trap to the condenser.

(*n*) *Application to operation with exhaust steam.*—The orifice method may also be applied to advantage to the operation of evaporators with auxiliary exhaust. The flow of steam through an orifice operated on exhaust steam at a pressure of 15 pounds gage will be constant in amount as long as the pressure on the discharge side is 2.5 pounds gage or less. With this for the maximum coil pressure, the shell will need to be operated under a vacuum. With a vacuum of say 26 inches, a temperature difference of 100 degrees is available. With clean coils this temperature difference is not required, and in single effect operation would cause serious priming when operating by the usual method. With an orifice installed, the capacity will remain constant regardless of vacuum or temperature difference until the coil pressure reaches 2.5 pounds gage. The operation need not be stopped here, but can be continued until the coil pressure has increased to 3 or 4 pounds gage with only a slight decrease in capacity. At this point the by-pass may be opened and the operation continued with the full exhaust pressure of 15 pounds on the coils without producing excess capacity with the consequent dangers of priming. The value of the method is that upon starting with clean coils a vacuum may be carried on the shell and the proper water level may be carried without danger of the priming which would otherwise result. In fact, all the inherent advantages of the orifice method are realized.

(*o*) *Advantages of method.*—The advantages of the orifice method may be summarized as follows:

(*a*) Any predetermined capacity, within the limit of capacity of the evaporator, may be obtained and maintained.

(*b*) The liability of priming is reduced to a minimum.

(*c*) Higher water levels may be safely carried at all times without danger of priming, and, for this reason,

(*d*) The scale forms on the coils in a thin uniform layer, and may be easily removed.

(*e*) The best operating conditions may be easily determined, and, on account of the constant capacity, may be easily maintained. At the same time, variations from the desired operating conditions, within wide limits, do not affect the capacity.

(*f*) The reducing valve is dispensed with. The evaporator is operated directly on the line pressure, and no pressure regulation is required. There is no danger of sudden increases of coil or shell pressures.

(*g*) The pressure on the inlet side of the orifice indicates exactly the amount of steam being used by the evaporator. The pressure on the coils indicates the condition of the coils as regards scale and heat-transfer coefficient.

(*h*) The system may be easily applied to any existing evaporating plant, single or multiple effect, high or low pressure.

It is believed that the adaption of the orifice method to the operation of evaporators will remove from the evaporating plant much of the mystery and uncertainty with which its operation is at present associated, and will place the evaporators upon as safe and sound a basis as have any of the ship's auxiliaries.

Acknowledgments are due to Rear Admirals C. W. Dyson and T. W. Kinkaid and to Mr. W. L. DeBaufre for encouragement and suggestions, and to Mr. Leslie E. Sebald for valuable assistance in the conduct of tests and the preparation of this article.

# LUBRICATION AND LUBRICANTS.

By G. R. Rowland.*

"Lubrication" and "Lubricants" may be defined as follows:

Good lubricants are substances, semi-fluid or fluid, capable of forming and maintaining films of sufficient thickness between two rubbing surfaces to prevent actual friction between the surfaces, substituting for it the fluid friction of the lubricant itself.

Lubricants in marine practice may be classed as mineral oils, animal oils, vegetable oils and greases.

Mineral oils are manufactured from crude petroleum and shale. The petroleum lubricating oils used in the United States are made from crude oils which may be classed in the following manner according to their natural characteristics:

*Paraffine Oils*—Composed chiefly of hydro-carbons of the paraffine series and found chiefly in Pennsylvania, West Virginia and Eastern Ohio.

*Asphaltic Oils*—Composed of hydro-carbons of the napthene series, found principally in California, Texas, Louisiana, and the Mid-Continent Oils produced in Kansas, Oklahoma and Illinois.

Lubricating oils made from the paraffine crudes have high gravities, high flash and fire tests, high congealing points and low viscosities unless blended with steam-cylinder oils. Asphaltic base oils have low gravities, lower flash and fire tests, congeal only at temperatures below zero degrees F., and can be made in any desired body without blending.

Lubrication can perhaps be best understood by considering the case of a spindle revolving at a speed of 10,000 revolutions

*Supervising Engineer, The Texas Company.

per minute in a bath of oil. The oil film immediately touching the surface of the spindle will revolve in the bath at nearly the surface speed of the spindle, the oil clinging to the spindle through its power of adhesiveness. The next succeeding films or layers of oil revolve at slightly less speed, while the film touching the sides of the bath will be as nearly stationary as its cohesiveness and the general movement of the oil will allow. The friction of the particles of oil sliding past one another is referred to as internal or fluid friction.

If a gradual side pressure is applied to the spindle, forcing it towards one side of the bath, some of the oil will be pressed out, and when the pressure becomes great enough the metal surfaces of the spindle and side of the bath or base will actually come together. This will produce heat and wear from metal friction. If the pressure is constantly kept at the point where the metal surfaces actually come together, an oil having more body or viscosity should be used. The heavier-bodied oil will resist the pressure and keep the surfaces apart, and the spindle will again turn in the center of the bath. As long as the oil is heavy enough to keep the metal apart no metal wear takes place. Again increasing the side pressure on the spindle will force the heavier oil out, and heating will again occur. To overcome a steady pressure of this nature a still heavier-bodied oil must be used in order to keep the surfaces apart.

In the above example the frictional heat of the spindle base will be slightly increased as the heavier oil is used under heavier pressure, but in no case will the heat be as high as that resulting from the destructive abrasion of metal on metal due to the use of too light an oil. Also, the minimum amount of power will be required to operate the spindle with the lightest oil in use, and the power required will slightly increase as the heavier oil is used; but, as in the case of temperature, the power will at no time be as high as when the surfaces are together through too great pressure or too light an oil. This case illustrates the effect of pressure upon the oil film and upon

the viscosity or body of the oil. The same rules apply generally to all bearing lubrication.

Lubricating oils can be divided into two general classes. The first class, called engine and machine oils, including dynamo, turbine, spindle oils, etc., is intended for external lubrication. The second class, steam-cylinder and valve oils, is used for the internal lubrication of steam cylinders and valves. Motor oils and oils for internal-combustion engines may be considered as a separate class, though they are not essentially different from some of the highest class turbine, engine and machine oils.

One cause of loss of power in a steam engine is friction. In all engines there are so many moving parts that it is of the greatest importance that friction should be reduced as much as possible. This is done by making the surfaces in contact smooth and of ample size; also making them of different metals and the use of suitable lubricants. Most of the marine engines in use at the present time are vertical multiple-cylinder machines built without fly-wheels or governors. Up until about twenty years ago the moving parts of marine engines were lubricated with various kinds of animal oils, principally lard oil. Of late years, or since the introduction of mineral oils, animal oils have been almost completely eliminated due to the fact there has been a vast improvement in mineral oils and also that lubrication is today done on a more efficient basis. In most all reciprocating engines the oil is reclaimed, filtered and used over again, which was impossible to accomplish to any great extent with animal oils.

Engine and machine oils should meet the following requirements:

(1) The oil should be of sufficient viscosity or body to keep the bearing surfaces apart at working temperatures.

(2) It should possess such qualities as will reduce frictional losses to a minimum.

(3) It should remain fluid at such low temperatures as will be met under ordinary service conditions.

(4) It should meet service requirements as to durability, separation from water, etc.

(5) The flash point should be sufficiently above the working temperature to allow a reasonable margin of safety.

(6) It should have no tendency to decompose or to form deposits which will gum up the machine and increase the friction.

(7) It should contain no impurities which will corrode or pit the bearing metal.

In general, but more especially with high and medium-speed machines, the best practical guide as to the suitability of a lubricating oil is the amount of frictional heat produced with that oil in use. Frictional heat indicates wasted power, and, in the case of a bearing, is determined by the difference between the temperature of the bearing in operation and the temperature of the room in which the machine is running. In general, one of the indications of an improvement in lubrication is the reduction of the amount of frictional heat. However, there are so many other factors which enter into the determination of the ultimate value of a lubricant that the amount of frictional heat as indicated by the bearing temperature is not in itself conclusive proof of the superiority of one lubricant over another.

However, the military safety of a battleship is of greater importance than a slight reduction in the temperature of a turbine bearing. A hot bearing or a burnt-out bearing at a critical moment might mean the loss of the ship, whereas the undue heating could have been prevented by the use of an oil of sufficiently high viscosity. In the case of Diesel engine lubrication, due to the fact that the unused cylinder oil drains out and mixes with the oil in the general lubricating system, it is desirable to use one oil sufficiently heavy for use in the cylinders, though this oil might cause additional internal friction in the bearings.

Motor oils and oils for internal-combustion engines are sub-

American Society of Naval Engineers



jected to extraordinary conditions. Their requirements are therefore somewhat different from other lubricating oils, and may be classed as follows:

(1) The viscosity should be such that the film of oil will be sufficient to keep the metal surfaces apart.

(2) The oil should possess such characteristics as will permit it to form a seal between the piston rings and cylinder walls so as to secure a high degree of compression.

(3) It should be of such a nature as to be converted into vapor without decomposition, even under extreme heat, so that the oil which gets into the combustion chamber will pass off with the exhaust as vapor without leaving an excessive carbon deposit.

(4) It should be of such a nature that whatever carbon is deposited is soft, and easily removed.

(5) The cold test should be low enough to meet all temperature conditions.

The proper viscosity for a motor oil should be determined by the piston-ring clearance and the conditions of the rings and cylinder walls. A wide clearance or worn rings, either of which causes poor compression, requires a high viscosity oil in order to secure the best results. If the piston rings are closely fitted to the cylinders a light-bodied oil should be used. If too light an oil is used too much oil will get past the piston rings on the suction stroke, and on the compression stroke some of the fuel mixture will leak past the piston rings and condense in the crank case, making the oil still thinner. If the oil is too viscous there will be a loss of power. Good compression depends very largely upon proper lubrication.

Mechanical conditions involved in steam-cylinder lubrication require special oils for that purpose. In color cylinder oils are usually dark green or black, though the filtered oils are light green and bright red. Steam cylinder oils are, as a rule, very viscous, some of them scarcely flowing at ordinary temperatures. The viscosity of these oils is taken at 210

degrees F., whereas that of ordinary lubricating oils is usually taken at 100 degrees F.

The conditions which are to be met in the steam cylinder are quite different from those occurring in the lubrication of external parts. In the case of steam-cylinder lubrication the oil is broken up or atomized and used only once and is then carried off with the exhaust steam. The chief considerations in the selection of a steam-cylinder oil should be, first, its viscosity, and, second, whether or not it should be compounded.

The speed and type of the engine, the method of effecting the lubrication, the distance from the steam chest to the point where the lubricant enters the steam line, and the amount of moisture in the steam should all be taken into consideration in determining the proper viscosity for a given case. It is necessary that the oil should be atomized or broken up into minute parts before it reaches the valves so that it will be carried by the steam and spread over the valves and cylinder walls in a finely-divided state. Otherwise, the oil will pass out of the exhaust in nearly the same condition in which it entered the steam pipe, without providing any effective internal lubrication. A light-viscosity oil atomizes more readily than a heavy-viscosity oil, so that the greater the amount of moisture in the steam the lower the viscosity of the oil should be. The shorter the distance which the oil has to travel in the steam pipe, the less the opportunity for atomization, and therefore the lower the viscosity of the oil should be. On the other hand, the farther away from the steam chest the oil is put into the steam pipe the heavier it can be, as time and distance are required for the heavier oil to properly atomize.

In ordinary cases, where there is a considerable amount of moisture in the steam, the cylinder oil should be compounded with a certain amount of animal oil so that the oil will more readily form an emulsion and adhere to the cylinder walls. On the other hand, a straight mineral cylinder oil should be used where it is desirable to keep the animal matter out of the

exhaust steam, as for example when the steam is condensed and returned to the boilers and used for other purposes.

## FIXED OILS AND COMPOUNDED OILS.

Before the introduction of mineral oils for steam cylinders tallow was used exclusively, being melted in a tallow pot which was kept on top of the steam chest. After the tallow was properly melted, it was introduced into the cylinder through a tallow cup, usually fitted with a valve and screw cover, and located on top of the steam chest. Lubrication by this method was crude and very intermittent. However, it probably suited the times and the condition of the engines, examples of which can be seen and studied with interest in some of our great museums. Sperm oil was formerly used, both mixed and unmixed, with mineral oils, for light machinery and spindles. Porpoise-jaw oil is still used for the lubrication of watches and delicate mechanism, for which a thoroughly non-drying and fairly limpid oil having an extremely long life is required.

As far as the lubricating properties of animal and vegetable oils are concerned, they were excellent, as the heavier oils, such as castor, were able to withstand very high pressure, and the lighter oils, such as sperm, had a very low coefficient of friction. The great drawback to their use, in addition to their high cost, was their oxidizing tendency, which necessitated the frequent cleaning of machinery to get off the gum formed by this oxidation. Moreover, acids were formed, which attacked the metals, causing pitting and corrosion. Therefore, pure mineral lubricating oils, if properly manufactured, can be used over and over again, which is not the case with any of the fixed oils.

Nearly all lubrication of a general nature can now best be cared for by a straight mineral oil of the proper viscosity. For turbines or for any circulating system, a compounded oil is never used, as it causes an emulsion, which above all is to be

avoided, and only the best grades of straight mineral filtered oils should be used for the latter purpose and should be selected strictly upon the basis of quality and suitable viscosity.

## GREASES.

In general lubricating practice there are certain mechanical conditions which cannot be handled to advantage by lubricating oils, and for which it is necessary or desirable to use a semi-solid lubricant. To meet such conditions the form of the mineral oil is changed to a semi-solid state by a special process, and the resulting lubricant is known as grease.

Grease, as the term is used for commercial purposes, is a mineral oil thickened with soap. The soaps used for this purpose are made from an animal or vegetable oil, such as tallow oil, horse fat, lard oil, cottonseed oil, rapeseed oil, palm oil and rosin oil. The consistency of the grease is from soft to hard, depending upon the percentage of soap in the mixture. In practically all greases mineral oil constitutes by far the larger part, frequently as much as ninety per cent of the finished product.

A grease manufacturer must be very expert in the methods of handling the various materials used in the manufacture of grease. Two greases might be made according to the same formula and still vary greatly in lubricating value, due to the difference in the methods of handling the various products used in their manufacture.

In the manufacture of most greases the animal or vegetable oil is placed in a steam-jacketed kettle and heated to the required temperature. Cream of lime or other alkali dissolved in water is then added, and this mixture is boiled and stirred until it is completely saponified, or turned into soap. In the manufacture of the best greases the heating is continued until nearly all of the moisture is driven off. Part of the mineral oil is then stirred in, and the stirring is continued with the

addition of more oil until the proper consistency is reached. Lime and soda greases are made by first mixing the fixed oil and mineral oil and then adding the alkali. The finished product should contain no lumps and should be homogeneous in character.

For all lubrication where the speed is high and where oil can be retained in or around the bearing, it is inadvisable to use grease, for the reason that lower bearing temperatures and lower power consumption can be secured by the use of oil. There are many places, however, where it is necessary to use grease, due to the fact that oil cannot be retained on the bearing surfaces, or due to the fact that the bearings are so located that it is impossible to get at them to oil them regularly. It is therefore necessary to have a lubricant that will feed very slowly and insure lubrication for an extended period without replenishing, and as a general rule, grease should be used for lubrication only under slow speeds and high pressures, or in cases where fluid lubricants cannot be retained for a sufficient time or used with sufficient cleanliness during the period of operation.

The melting point of grease is the temperature at which the grease begins to flow in liquid form. There are several methods of determining the melting point and each method will give a different temperature as the melting point. The following method, Fig. 1, is now being used very generally with satisfactory results. Some of the grease to be tested is spread with the fingers onto the bulb *B* of a thermometer, which ranges from about 100 degrees F. to about 400 degrees F. The thermometer is then fitted by means of a cork, *C*, into an empty test tube *T*, so that the thermometer hangs free. The test tube is then placed in an oil bath *O* heated by a Bunsen burner, and the flame so regulated that the temperature will rise from two to four degrees per minute. When a drop forms on the bulb of the thermometer and falls to the

bottom of the test tube, the temperature is recorded as the melting point of the grease.

Cup greases are primarily intended for use in compression cups which force the grease directly to the bearing surfaces. There are, however, a number of designs of compression and other types of cups. As cup greases are comparatively low in melting point, very little frictional heat is required to reduce them to a liquid, consequently their consistency should depend very largely upon the mechanical method used to apply them to the bearing surfaces. The personal equation plays a large part in the selection of greases of various consistencies, as different engineers with the same equipment vary in their opinions, some preferring a very soft and others preferring a very hard grease, irrespective of the requirements of the machine.

Unless compression cups are used, grease must depend upon the frictional heat of the bearing to soften it enough to be carried in between the bearing surfaces. At the same time it is necessary that the grease should not melt and flow away from the bearings, but should stay in a semi-solid state under bearing temperatures. Special grease, intended for use in relatively high temperatures, can be of the same general consistency as the compression-cup greases, but of higher melting point. This permits of direct application to the bearings, as in the case of open-slot bearings, or it can be fed through a copper pin cup. This cup does not have the compression device, but instead has a copper wire, the function of which is to transmit heat from the bearing to the grease, so as to make it fluid enough to flow to the bearing. For extremely high surrounding temperatures a very hard and a very high melting point grease should be used.

If a grease is not properly manufactured, the oil will frequently separate under service conditions, leaving the soap in the cups and on the bearings in solid or powder form. This is caused by a defect in manufacture that cannot be discovered

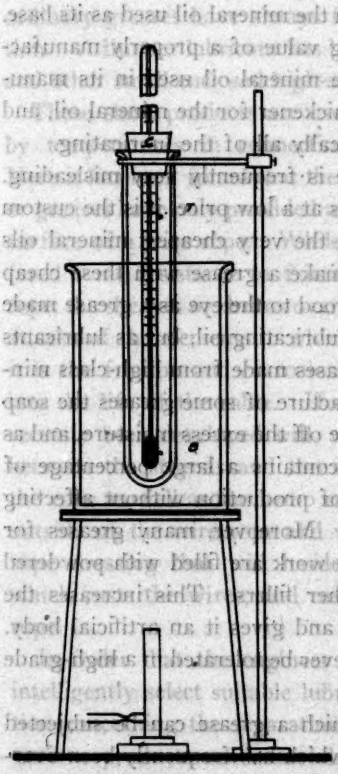


FIG. 1.

FIG. 2.

by the casual inspection of a sample of grease. Cup greases and general lubricating greases should not separate when subjected to working temperatures, and when cooled should go back into a perfectly homogeneous mixture.

A grease is never better than the mineral oil used as its base. In other words, the lubricating value of a properly manufactured grease depends upon the mineral oil used in its manufacture. The soap acts as a thickener for the mineral oil, and it is the latter that does practically all of the lubricating.

The appearance of a grease is frequently very misleading. In endeavoring to make greases at a low price, it is the custom of some manufacturers to use the very cheapest mineral oils obtainable. It is possible to make a grease with these cheap oils which will appear just as good to the eye as a grease made from the very best grades of lubricating oil, but as lubricants they are greatly inferior to greases made from high-class mineral lubricants. In the manufacture of some greases the soap is not boiled down so as to drive off the excess moisture, and as a result the finished product contains a large percentage of water. This lowers the cost of production without affecting the appearance of the grease. Moreover, many greases for heavy pressures and for rough work are filled with powdered soapstone, chalk, talc, and other fillers. This increases the weight, cheapens the product, and gives it an artificial body. Such fillers, however, should never be tolerated in a high-grade grease.

The low temperature to which a grease can be subjected without solidifying is a point which has frequently been overlooked in the past. It is particularly necessary that greases used in low temperatures should be manufactured from low cold-test oils, and therefore they will not become solid even in very severe winter weather.

There are certain conditions which the hardest and the highest melting point greases cannot satisfactorily meet, and include such work as that offered by different types of gears

and wire ropes. To meet these exact conditions a product has been perfected which, due to its great viscosity, its cohesiveness, its adhesiveness, and its heat resisting properties, is now being used with wonderful success under conditions where ordinary greases have always failed to show efficient lubrication. A thinner lubricant has been developed for worm drives and other small enclosed high-speed gears.

These same products are used under conditions as affected by temperatures in connection with wire-rope lubrication. Here it has been found that the ordinary or best grades of oils or greases have only provided a temporary coating of the outer surface of a wire rope. While the outer coating lasted it offered some resistance to friction between the surface of the rope, the sheaves or drums over which the rope passes or to the attacks of the elements to which the rope may be exposed, but it is the friction and corrosion of the inner strands that eats into the life of the cable. However, the lubricant which has recently been developed will, when properly applied, work in between the strands and thoroughly lubricate every wire of the rope as well as the hemp center; it will not harden on the outside of the wire rope and will not peel or crack, and any rope dressing that will harden or crack will let in water, which rusts the wires, and rust is the great enemy to a wire rope.

In order that the engineer may be in a position to more intelligently select suitable lubricants a little general information concerning the separation of the natural crude oils into various products used for lubrication purposes, and also by touching upon the most important test to which lubricating oils are submitted will give a better idea regarding the selection of proper lubricants, it being desired to bring out clearly those qualities which will cause an oil to be suitable or unsuitable for the contemplated work.

### OIL REFINING.

Briefly, crude oils are separated into various products by

the following methods of distillation—the "dry" distillation process in which the heat is applied directly to the bottom of the stills and the distillation continued until coke only remains and the "steam" process in which beside the direct heat steam is permitted to enter at the bottom and bubble up through the body of the oil, also the "vacuum" process in which, by means of a pump, a partial vacuum is created in the still. These various methods may be used singly or, as in most cases, two or more are combined before the finished products are arrived at. The advantage or disadvantage of using either or any of them would require too much space to properly cover here.

The stills, of course, vary greatly in design and size. In any case they are similar to an ordinary steam boiler without the tubes, usually being a horizontal steel shell set in brick work and connected at the top or dome with vapor pipes which lead to the condenser coils.

The boiling points of the various constituents of the crude of course determine when that part vaporizes and passes to the condenser.

The more volatile portions of the crude, such as benzine, gasoline, kerosene and gas oil are distilled first, the cut from which general lubricating oils are made coming over considerably later, and the steam-engine cylinder stock is the residue which remains in the still after the other products are drawn off, providing steam is used in distillation.

Each cut coming from the still may, and usually does, undergo, further distillation by different processes, or they may be acid treated, filtered through fullers earth, filter pressed sweated, or sun bleached, each process having its value in producing a product for a certain purpose. For instance, the cut from which general lubricating oils are made is subjected to numerous costly processes before it is ready for the consumer.

Filter-pressing and sweating are certain processes by which the paraffine wax is removed from the stock. Some oils are

sun-bleached or exposed in shallow pans to the sun, which causes them to have a permanent color.

Paraffine-base lubricating oils when finished seldom have a viscosity of over 250 seconds Saybolt, hence, in order to obtain the higher viscosities which are necessary for certain purposes, they must be blended with some other product of higher viscosity. The residue from the Pennsylvania crude is usually selected for this purpose, although various animal or vegetable oils are sometimes used. Asphaltic-base oil can be finished at practically any viscosity and they thus avoid the objectionable necessity of blending.

Paraffine-base oils usually do not have a low cold or pour test, owing to the fact that it is both difficult and costly to remove the paraffine wax which is the cause of their freezing at low temperatures. They will seldom flow at a temperature lower than 30 degrees F. Asphaltic-base oils, not having to contend with the paraffine wax are practically zero cold-test oils, that is, they will still flow at zero. Many of them will not freeze and will still flow at 15 degrees below zero.

This quality is very important when an oil is to be used in or on equipment subjected to extremes of temperatures.

### OTHER OILS.

Animal, fish and vegetable oils are sometimes used, either alone or, as usually is the case, as a compounding for mineral oils for different lubricating purposes. The more important are as follows:

Whale Oil.....made from whale "blubber,"
Lard Oil.....made from the fat of hogs,
Neats Foot Oil...made from beeves' hoofs,
Tallow Oil.....pressed from beef tallow,
Sperm Oil.....made from the head of the sperm whale,
Cottonseed Oil...made from cotton seed,
Rapeseed Oil...pressed from rapeseed,

Castor Oil. . . . . made from the castor bean,
Rosin Oil. . . . . made by destructive distillation.

Certain of the above, such as lard and tallow oils, not only have value when compounded with mineral oils, but for some purposes good results can be obtained only by so compounding.

### SELECTION OF AN OIL.

It can be safely said that the only way to see what a lubricant is is to actually try it out on the equipment that is to use it; however, it is not always possible to follow this method, and also, even after a certain lubricant has been proven and selected, it is sometimes desirable to have certain specifications as a guide in future purchases. This can only be accomplished through a series of laboratory tests, and the following is a list of those tests generally made. They, of course, need not all be made in selecting oils for certain purposes:

1. Gravity,
2. Viscosity,
3. Color—Odor,
4. Cold Test,
5. Flash,
6. Fire,
7. Carbon,
8. Water and Sediment,
9. Emulsification Tests.

Each of the above tests will be considered in the following with cuts of necessary instruments for making such tests:

*Gravity.*—By the gravity of an oil is meant its relative weight as compared to that of distilled water. Since a given volume of distilled water at a certain temperature always has the same weight, the weight of the oil to be tested, instead of being expressed in grams or pounds, is said to have a weight which is a certain decimal fraction of the weight of the same

volume of distilled water at 60 degrees Fahrenheit. This decimal fraction is known as the specific gravity of the oil, which may be defined as the ratio of its weight to the weight of an equal volume of distilled water. For example, if a certain volume of oil weighs 21 grams and the same volume of water weighs 25 grams, then the oil weighs 21/25 or 0.840 as much as the water, and accordingly is said to have a specific gravity of 0.840.

Specific gravity can be determined in various ways, one of which is by means of the pyknometer, or specific-gravity bottle, Fig. 2. A special bottle with a perforated stopper is used in the laboratory, but with care an ordinary bottle can be used. The bottle should first be thoroughly cleaned and rinsed and then filled with cold distilled water and heated to 15.6 degrees C., or 60 degrees F. At this temperature the water should just fill the bottle. The full bottle should be carefully weighed, and after emptying and drying it, the empty bottle should be weighed. The bottle should then be filled with the oil to be tested, heated to 60 degrees F., again having the bottle exactly filled with oil and weighed. In each case the weight of the empty bottle must be deducted from the total weight found. The specific gravity of the oil is the ratio between the actual weight of the oil and the actual weight of the water.

To the oil manufacturer or chemist a knowledge of the gravity of an oil is important, and several other instruments are in use to make these determinations, but we have given the simplest means whereby the engineer may make some simple determinations should he so wish. In the first place it is one indication of the crude from which the oil is made. Light-gravity crudes are entirely paraffine-base crudes, and products of such crudes are of correspondingly light gravity. The Pennsylvania, Ohio, Indiana, and northern Louisiana crudes are paraffine-base and light-gravity crudes. The Texas, except from the Electra field in the northern part and the California crudes, are heavy-gravity, and the products made from

these crudes are correspondingly heavy in gravity. The experienced oil man can locate with a reasonable degree of accuracy the source from which a certain oil came, and in so doing its gravity is his chief guide. Furthermore, in refining and handling a certain crude, gravity is an important factor, as it is a guide in the distillation process.

It will be remembered that the value of gasoline is determined by its range of distillation and not by its gravity. Of two gasolines having the same range of distillation, the heavier will give the greater power and the greater number of miles per gallon, due to the greater number of heat units. Gravity has no bearing upon the value of kerosene. In the case of lubricating oil, the gravity is of value in that it is an indication of the nature of the crude from which the oil is made. In the case of fuel oil, gravity is of more importance as it is an indication of the heat units in the fuel, the heavier the gravity the greater the number of heat units per gallon.

In selecting a lubricating oil, the oil expert never takes gravity into consideration, as it has nothing to do with the lubricating value. Twenty-five years ago all American oils came from Pennsylvania and the North and were of light gravity. When the heavy-gravity crudes first appeared, though very valuable, they were entirely different; the refiners evidently did not know how to handle them to the best advantage, and for a time the products were the results of practical experimentation on a large scale.

At the present time, however, for many purposes much better lubricating oils are being made out of these heavy-gravity crudes than were even dreamed of five years ago. Properly manufactured oils from the southwestern heavy-gravity crudes have certain natural advantages for some purposes.

Specifications issued by most buyers of oils have now been widened to include oils made from all crudes. For instance, a specification written ten years ago for a gear oil would call

for 25 degrees B. to 26 degrees B. gravity; today, to include the best gear lubricants, the specifications must call for 13 degrees B. to 26 degrees B, which is so wide as to be nearly no specification at all, yet this is exactly the change that is taking place.

Lubricating products are today made from such a wide range of crudes and therefore are of both light and heavy gravity. So far as the consumer is concerned, however, gravity has very little significance, except possibly in the case of fuel oil.

*Flash and Fire.*—When any liquid is heated sufficiently, it gives off vapor. For example, as water is heated it is converted into steam, gradually until the boiling point is reached, and then rapidly. The same phenomenon occurs in the case of oil, but in this instance the vapors which are given off are inflammable, and form an explosive mixture with the air. The danger of fire or explosion, especially in the case of kerosene or burning oils, led to the introduction of a test which would differentiate those oils which were safe for illuminating purposes from those which were so volatile as to make them dangerous. Later, when the mineral lubricating oils were introduced, the same tests were extended to these heavier oils, not because there was any particular danger of fire in the case of a properly manufactured oil, but rather as a guide to the refiner in his various methods of distillation.

The flash point of an oil is the temperature at which an oil, upon being slowly heated, will give off sufficient vapor to form over its surface a mixture with the air which, upon the introduction of a small flame, will ignite and go out. As the temperature of the oil rises, more vapor is given off, and when the production of vapor is sufficiently rapid, the oil takes fire and burns with a continuous flame. The fire point of an oil may therefore be defined as the temperature at which, upon the application of a flame, the oil will burn continuously.

For the determination of the flash and fire points a number of instruments have been devised, which, with but few exceptions, can be divided into two general classes open cup and closed cup. In the first case the sample is contained in an open reservoir, having free access to the air, and the flash point is the temperature at which sufficient vapor is generated over the immediate surface of the oil to form an inflammable mixture. In the second case the sample is held in a container partially or wholly closed, with, however, sufficient air space to allow the formation of an inflammable mixture of air and vapor. As can readily be seen, the flash point of the same oil will be lower when taken with a closed cup than with an open cup. In order to ignite, it is necessary that there should be a mixture of a certain amount of vapor with a certain amount of air. As the quantity of air is smaller in the closed cup, it will not require the formation of as much vapor to give this mixture; consequently a lower flash point will be recorded when the closed cup is used.

The open cups are best adapted to ordinary work and, if used with care, give very satisfactory results. Of these the Tagliabue and Cleveland open testers are the most important, cuts of which are herewith shown. There are a large number of other testing cups which space will not permit us to dwell upon.

Flash and fire tests are of special importance in the case of kerosene and petroleum spirits. Burning oils going into different States must be tested with the instruments which these States have legally adopted. While some of the States require no special cups or require only "standard instruments," others specify the Tagliabue, the Foster, or the Elliott instrument. Most of the State laws specifying a minimum flash or fire test for burning oils were drawn up at a time when kerosene was the most valuable constituent of petroleum and when gasoline was a drug on the market. At that time refiners endeavored to increase their yield of kerosense by lowering the

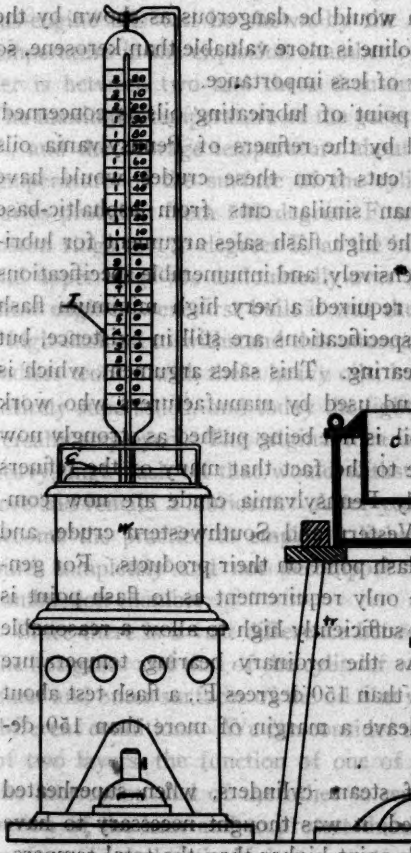


FIG. 3.—TAGLIABUE OPEN TESTER.

FIG. 4.—CLEVELAND OPEN TESTER.

flash point, with the danger of making a product which had too great a fire hazard, and was dangerous for domestic use. Consequently State laws were devised to prohibit the sale of any burning oils which would be dangerous as shown by the flash point. Today gasoline is more valuable than kerosene, so that these laws are now of less importance.

As far as the flash point of lubricating oils is concerned, it was soon discovered by the refiners of Pennsylvania oils that given lubricating cuts from these crudes would have higher flash points than similar cuts from asphaltic-base crudes. Accordingly, the high flash sales argument for lubricants was used very extensively, and innumerable specifications were drawn up which required a very high minimum flash point. Some of these specifications are still in existence, but they are rapidly disappearing. This sales argument, which is a talking point only, and used by manufacturers who work only on paraffine base oil, is not being pushed as strongly now as at former times, due to the fact that many of the refiners who formerly used only Pennsylvania crude are now compelled to draw from Western and Southwestern crude, and therefore to lower the flash point on their products. For general lubricating oils the only requirement as to flash point is that the flash should be sufficiently high to allow a reasonable margin for safety. As the ordinary bearing temperature would not average more than 150 degrees F., a flash test about 300 degrees F. would leave a margin of more than 150 degrees.

In the lubrication of steam cylinders, when superheated steam was first employed, it was thought necessary to have cylinder oils with a flash point higher than the total temperature of the steam. Superheated steam in some instances reaches a temperature as high as 700 degrees F. It is possible to have a cylinder oil made from certain stocks with a flash point of nearly 620 degrees F., but these oils are so near the end of the distillation that it was found in practice with high

superheat temperatures that the oil that stayed in the cylinder changed very quickly to coke and caused great damage.

The fallacy of the high-flash argument on lubricants for gas-engine oils can be shown by the following facts. The temperature in the explosion chamber of the gas engine cylinder is between two and three thousand degrees Fahrenheit. The maximum temperature of the gases is about 2,700 degrees F., and the average temperature about 950 degrees F. The temperature of the surface of the cylinder walls themselves varies probably from 500 degrees F. at the top travel of the piston rings to 175 degrees F. at the bottom. The piston top and upper rings are undoubtedly over 500 degrees F. When these engines were first built it was thought necessary to use a high-flash oil to withstand this unusual condition. Owing to endless trouble with these heavy oils, however, they have been entirely superseded, and today the lighter oils are almost universally employed. The original idea was to get, if possible, a heavy, high-flash oil that would withstand the high temperature and remain a lubricant. After years of experience it has become the practice to employ a lighter bodied oil that will burn completely and not decompose when it gets into the combustion chamber.

Since there is a vast difference between the temperature of the hot gases and that of the cylinder walls, it follows that the inner and outer surfaces of the oil film will be exposed to quite different conditions. We can consider the oil film as consisting of two layers, the function of one of these being to furnish the lubrication and of the other to withstand the destructive action of the heat and to protect this lubricating layer. The surface exposed to the high temperature of these gases is without doubt affected by this heat, irrespective of its flash point. The outer surface is exposed only to the comparatively low temperatures of the cylinder walls, and, with a film of any appreciable thickness, will be protected from the heat of the burning gases.

Experiments which we have made show that the flash point has nothing to do with the lubricating value of a motor oil. An examination of the flash point of the oil in the crank case, after it had operated in the motor of a car that had run fifty miles, revealed the fact that the flash was only 150 degrees F., while the original flash point was 445 degrees F. The original high flash of the oil did not indicate the sealing qualities necessary for a good motor oil, and with this oil the seal between the piston rings and the cylinder walls was not such as to prevent the gasoline vapor from passing into the crank case during the compression stroke. This gasoline mixture contaminated the oil, made it lighter in body and only 150 degrees F. in flash, thus proving that a high-flash oil would not remain so unless it possessed other and more valuable characteristics, and that an oil having these essential qualifications could well be of a lower initial flash point if its flash would remain constant.

A great many purchasers of steam-cylinder oils, motor oils, and gas-engine oils, as well as many other lubricating oils, think that they require a lubricant having a high flash test. It has been proven in scores of cases that when it comes to practical results that for certain purposes certain low flash point oils are greatly superior to the high flash oils; so much so that among those well informed, the flash and fire points of properly manufactured oils are considered of practically no importance in determining their lubricating value.

*Color.*—For ordinary purposes the color of an oil can be determined by its appearance in an ordinary sample bottle. Color as well as the general appearance of an oil will vary with the size of the bottle, and it is therefore essential that for comparative purposes the same sized bottle be used. The four-ounce bottle is the standard used by all oil companies, and is therefore the most convenient to use.

In the laboratory, where color must be accurately determined, special instruments are used, the most important being the Lovibond Tintometer, Fig. 5. This consists of a covered

trough or box *T* divided longitudinally by a partition terminating in a vertical knife edge *K* opposite an eyepiece *E* fixed at one end. Two channels *L* are thus formed, which diverge slightly from the end at which the eyepiece is placed. The oil to be examined is placed in a rectangular glass cell *C*, which is placed at the end of the right-hand channel. Stained-glass slides of known depth of color are then inserted into the small

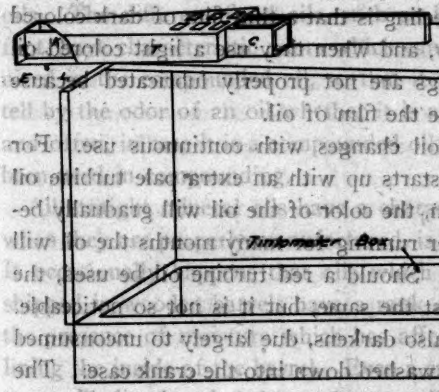


FIG. 5.—LOVIBOND TINTOMETER.

slots at the other end of the channel until the color of the oil is exactly matched. These slides are all numbered on a scale known as the five hundred series, and the depth of color of the oil is expressed numerically. For dark oils a half-inch cell is used, for pale oils a six-inch cell, and for kerosene and gasoline an eighteen-inch cell.

In general, color is no guide as to the suitability of any class of lubricating oil for any particular work. Some pale oils are better lubricants for some purposes than certain red oils, and, on the other hand, there are red oils which for certain purposes are superior to some of the pale oils. A high-grade oil may be either pale or red, and, on the other hand, a lower

grade oil may be either pale or red. The oils of any one refiner can, as a rule, be classed generally by colors, but, owing to the considerable difference in crudes employed and to the manipulation of these crudes, the products of one refiner cannot be compared with those of another on the basis of color alone.

Among the users of oil considerable prejudice has always existed in regard to color. Engineers who are accustomed to dark colored oils regard the light colored oils with distrust. The reason for this feeling is that a thin film of dark colored oil can be seen clearly, and when they use a light colored oil they think the bearings are not properly lubricated because they cannot readily see the film of oil.

The color of any oil changes with continuous use. For instance, if a turbine starts up with an extra pale turbine oil in a circulating system, the color of the oil will gradually become darker, and after running for many months the oil will be a deep wine color. Should a red turbine oil be used, the change takes place just the same, but it is not so noticeable. In Diesel engines, oil also darkens, due largely to unconsumed parts of the fuel being washed down into the crank case. The change in the color does not affect the lubricating properties of the oil provided it is properly filtered by a special process which can be easily and cheaply installed.

When an oil changes in color through use, it does not mean that the oil is becoming worn out or that its lubricating value is being materially affected. It is simply an indication that under the stress of service the oil is taking up various foreign particles which affect the color. Changes in color of an oil, therefore, are due to surrounding conditions and not to the oil itself.

In the case of cylinder oils, also, choice of color in the majority of cases is largely a matter of prejudice. Some engineers prefer a dark cylinder oil, others prefer the bright red filtered cylinder oil. Unless the exhaust steam is to be

used for manufacturing purposes where the oil may get into the articles being manufactured, conditions which can be met by the filtered cylinder oils can be met as well by a dark colored cylinder oil and vice versa.

*Odor* is a matter of importance both in the case of refined oils and in the case of lubricating oils.

Lubricating oils are classified in regard to odor as normal and compounded. Normal neutral oils are straight mineral oils, and compounded oils are mineral oils combined with fixed oils. The different fixed oils, such as lard, rape and neats-foot, have characteristic odors which can frequently be recognized in the compounded oil, so that it is usually possible to tell by the odor of an oil whether it is a straight mineral oil, and often, in case it is a compounded oil, what fixed oils have been used in compounding.

All straight mineral oils have a characteristic odor, which, when they are properly manufactured, is almost imperceptible. It occasionally happens that oils which have been stored or shipped in wooden barrels have a rank odor. This is due to the presence of moisture which has affected the glue used in lining the inside of the barrel. Even a small amount of moisture will dissolve the glue and cause the rancid odor. Due to modern facilities, inspection and care, it is practically impossible for moisture to get into the oil at the filling plant. Water, at times, however, does get into the barrels during transportation, and because the importance of keeping the oil barrels dry at all times is not properly understood, the barrels are frequently allowed to stand on end in the open. Snow or water accumulates on the heads, and a small amount will sometimes leak through, causing damage to the oil. To prevent water from getting into the oil in this way, barrels should be stored on the side or bilge; better still, they should never be allowed to stand in the open.

*Cold Test.*—Nearly all oils are liquid at ordinary temperatures. In the subject of "gravity" reference was made to the

fact that oils expand with an increase in temperature and contract with a decrease in temperature. If the temperature is gradually reduced from 60 degrees F. the oil continues to contract, and finally a point is reached at which it ceases to flow. Oils made from paraffine-base crudes, which contain paraffine wax in solution at ordinary temperatures, cease flowing at higher temperatures than oils made from asphaltic-base crudes. This is due to the fact that wax itself is solid at ordinary temperatures and will, when the oil is cooled, solidify very rapidly so that the oil itself cannot flow.

There are two changes which may occur in an oil when it is cooled, designated as cloud test and pour test. The cloud test is the temperature at which the oil upon being cooled becomes slightly cloudy or partially opaque from the formation of particles of paraffine wax held in suspension by the oil. This test is made only in the case of paraffine-base oils, as asphaltic-base oils contain no paraffine wax and therefore have no cloud test. The pour test is the lowest temperature at which the oil will flow or pour. This test is made on both paraffine and asphaltic-base oils. The term cold test is frequently used loosely to refer to either cloud test or pour test, more frequently to the latter.

There is a fundamental difference between paraffine-base and asphaltic-base oils. In the distillation of a paraffine-base oil, the wax, which is held in suspension by the oil, is a volatile product and is distilled over with the various petroleum distillates, and the residue left in the still after the distillation is complete is petroleum coke. In the case of asphaltic-base oils the asphalt is not volatile and cannot be distilled, and therefore does not distill over with the petroleum products, but remains as a residue in the still. Therefore, while a paraffine-base oil contains paraffine wax in solution, an asphaltic-base lubricating oil does not contain any trace of asphalt.

In making the cloud test, Fig. 6, a cylinder or glass jar about one and one-fourth inches internal diameter and about

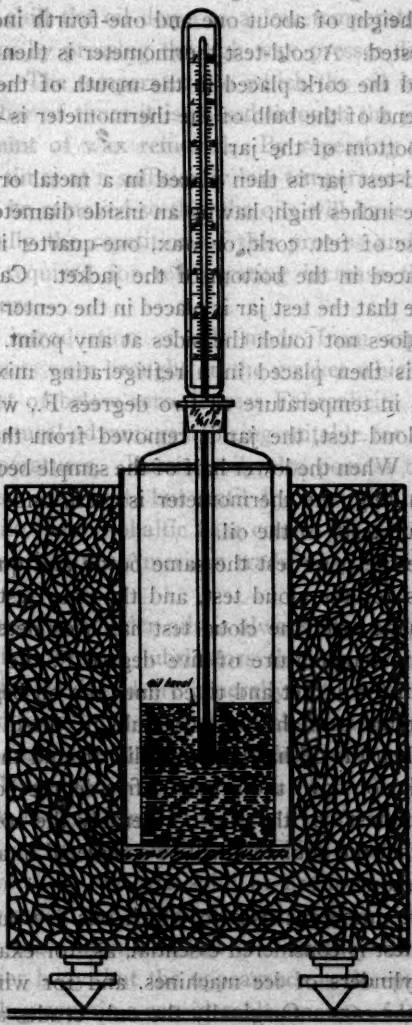


FIG. 6.

five inches long, or a regular four-ounce sample bottle, is filled to a height of about one and one-fourth inches with the oil to be tested. A cold-test thermometer is then inserted into a cork, and the cork placed in the mouth of the jar, so that the lower end of the bulb of the thermometer is one-half inch from the bottom of the jar.

The cold-test jar is then placed in a metal or glass jacket four to five inches high, having an inside diameter of the test jar. A disc of felt, cork, or wax, one-quarter inch in thickness, is placed in the bottom of the jacket. Care should be taken to see that the test jar is placed in the center of the jacket so that it does not touch the sides at any point. The whole apparatus is then placed in a refrigerating mixture and at every drop in temperature of two degrees F., when near the expected cloud test, the jar is removed from the jacket and inspected. When the lower half of the sample becomes opaque through chilling, the thermometer is read, and this is taken as the "cloud test" of the oil.

To make the pour test the same bottle and quantity of oil are used as for the cloud test, and the pour test may, if desired, be made after the cloud test has been determined. At each drop in temperature of five degree F. the bottle is removed from the jacket and tilted until the oil begins to flow. In the extreme case the bottle should be tilted to the horizontal. When the oil has become solid around the thermometer and will not flow, the previous five-degree point is taken as the "pour test" of the oil. Preferably the cold should be applied so that the pour test will be completed in approximately one-half hour.

There are numerous uses to which oils are put for which a low cold test is considered essential, as, for example, in the ammonia cylinders of ice machines, and for winter lubrication of motor cars. Originally the only crudes available for the manufacture of lubricating oils were the Pennsylvania crudes, which have a paraffine base and which contain paraf-

fine wax in solution. To free the oil from this wax it is necessary to chill the oil down to a low temperature and remove the wax by pressing through wax presses designed for this purpose. The temperature to which the oil is congealed and the number of times it is passed through the presses control the amount of wax removed. By repeating the process a number of times at a sufficiently low temperature enough of the wax will be removed so that the oils will have a zero cold test. Naturally the repetition of this process and the lower temperature required for refrigerating increase the expense of manufacture.

With the introduction of oils made from asphaltic-base crudes, products were brought on the market which had a natural cold test of below zero degrees Fahrenheit. These oils possess the natural advantage of being suitable for all extreme climatic temperatures. Russian oils also possess this characteristic of low cold test, but until the manufacture in this country of oils from asphaltic base crudes the amount of low cold test offered to the American trade was practically negligible.

The cold test of cylinder oils is always high. Cylinder oils are usually fed through lubricators that are heated or protected from the cold, and are generally not subjected to cold-weather conditions. No engineer thinks of trying to remove cylinder oil from the barrel until it has been standing in a warm room long enough to thin down. Cylinder oils are not usually made lower than 45 degrees F. cold test and some run as high as 80 degrees F. The high cold-test oils are usually filtered oils and are solid at ordinary temperatures, looking very much like a dark colored vaseline. Some engineers prefer these high cold-test oils, as they think the oils have a heavier body, but the appearance is deceptive for the reason that they contain a considerable amount of low melting-point wax, which makes them solid at ordinary temperatures, but with a slight amount of heat they become thin. All that

is necessary in cylinder oil is to have a sufficiently low cold test to allow it to flow through the lubricator, which may be exposed to drafts from windows or doors in the engine room. The conditions under which an oil is to be used determine whether or not a low cold test is essential. For general lubrication, where the oil is not exposed to low temperature conditions, a low cold test is unnecessary; on the other hand, where the oil is subjected to such conditions, a low cold test is desirable and, in some cases, necessary.

*Viscosity.*—In determining the suitability of a lubricating oil for any particular purpose, the most important single characteristic to be considered is its vicosity. In discussing the subjects of color, gravity, flash and fire, and cold test, it has been seen that color and gravity have nothing, and flash and fire practically nothing, to do with the lubricating value of an oil. While the cold test of an oil has an important bearing upon its suitability for feeding uniformly under low temperatures, generally speaking it has nothing to do with the lubricating value, once the oil has reached the bearing surface. Viscosity, however, is a matter of the greatest importance, and should always be taken into consideration, and should, therefore, be thoroughly understood.

The viscosity of an oil is the resistance offered by its particles or molecules in sliding past one another; in other words, its internal or fluid friction. Viscosity denotes the body or relative fluidity of the oil, and is determined by the time it takes a certain amount of oil to flow through a standard orifice. Glycerine and castor oil are very viscous or high in viscosity, and consequently flow slowly, while gasoline and water are low in viscosity and flow more rapidly.

The relative viscosity at atmospheric temperature of two or more oils in sample bottles can be roughly determined by holding the samples together and quickly turning upside down, and then noting the drop from the bottoms of the inverted bottles. The oil which drops first has the lower viscosity,

or the thinner body; if they drop together, the viscosity or body is about the same. Again, the samples may be shaken and the relative viscosity determined by watching the ascent of equal sized air bubbles, the more sluggish the movement the greater the viscosity of the oil.

For the accurate determination of viscosity it is necessary to use an instrument called a viscosimeter. The viscosimeter in general use in this country is the Saybolt Standard Universal. The standard instrument in England is the Redwood, in France the Barbey, in Russia the Lamansky-Nobel, and in Germany the Engler. There is also a number of other instruments in existence, as well as a number of modifications of the instruments already mentioned. Other instruments which have been used to a limited extent in this country are the Tagliabue, the Scott and the Doolittle.

Except for those interested in the export of oil, or those outside of the United States, the Saybolt Universal Viscosimeter is the only instrument with which the oil man needs to be familiar. At the present time there is a decided movement in this country toward its general adoption, so that it has the advantage of being the most generally used and generally known of any of the large number of viscosimeters. It is also simple in construction, is accurately standardized, and has a large bath, by which the oil can easily be kept at the required temperature. The heating arrangements are superior to those of any other viscosimeter on the market.

The Saybolt Universal Viscosimeter, Fig. 7, is made entirely of metal. The cylinder *C* which holds 83 c.c. (cubic centimeters) of oil, is surrounded by a large bath *B*. The upper edge of the cylinder is surrounded by a gallery *G*, into which any surplus oil flows. At the bottom of the cylinder is a jet *J*, through which the oil flows into the receiving flask *F*, which holds 60 c. c. to a mark on its neck. The jet is inclosed by the tube *T*, which extends below the orifice of the jet, and into the bottom of which a cork *L* with a string attached

is lightly inserted, so as to close the tube. The bath is provided with stirrers *S*, fitted with a thermometer for keeping the temperature uniform. The bath is heated by a gas-ring burner *R*, which extends around in a circle under the bath, or by a steam U-tube or by an electric attachment.

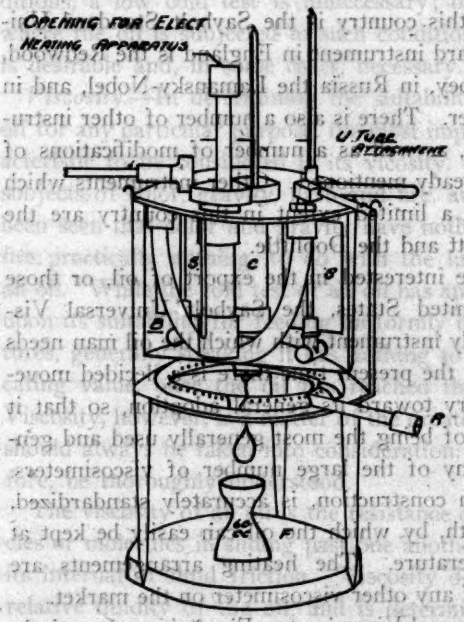



FIG. 7

To make the test the oil cylinder is filled with the oil to be tested, and the bath filled with water or oil which has been brought to the required temperature. The temperature of the oil being tested is adjusted by regulating the temperature of the bath. A thermometer is used to stir the oil in the cylinder, and when this oil is at the correct temperature and

the bath a trifle higher in temperature, the thermometer is withdrawn and the gallery emptied of all surplus oil by means of a pipette. This insures a positive starting point, as the oil will exactly fill the cylinder. The flow of the oil is then started by quickly pulling the string, which withdraws the cork *L*, and at the same time a stop watch is started and the time of the outflow of 60 c.c. recorded in seconds. The number of seconds required for the 60 c.c. of oil to pass through the orifice indicates the viscosity of the oil. All Saybolt instruments are so standardized that 60 c.c. of water at 60 degrees F will flow out in 30 seconds, and water accordingly has a viscosity of 30. A light-bodied spindle oil, the time of outflow of which is 75 seconds at 100 degrees F., is said to have a viscosity of 75 at 100 degrees F. A heavy machine oil might have a viscosity of 500 or even higher at 100 degrees F.

Numerous attempts have been made to find the relation between viscosities as determined by the Saybolt Universal, Redwood, Engler, and other viscosimeters. The difficulties are almost insurmountable, however, as the different instruments are standardized in entirely different ways.

The mistake is frequently made of thinking that an oil heavy in gravity is high in viscosity, and that an oil which is light in gravity is of light or low viscosity. The fallacy of this idea becomes apparent when one remembers that water is at the same time thinner in body and heavier in weight than any mineral lubricant. In the case of oils from the same crude, the oil heavier in body would generally be heavier in gravity, but between oils manufactured from different crudes there is no relation between viscosity and weight or gravity.

The temperature at which the viscosity test is made is a matter of the greatest importance, as all oils lose viscosity with a rise in temperature. The temperatures used for the Saybolt Universal are 210 degrees F. for steam-cylinder oils, and 100 degrees F. for engine and machine oils, and as a rule 100

degrees F. for internal-combustion lubricating oils. The temperature of the oil during the test must be kept uniform, as a variation of one degree in temperature would make an appreciable difference in the viscosity. Moreover, in referring to the viscosity of any oil the temperature at which the viscosity was taken should be stated.

In selecting an oil for any particular purpose the viscosity should be determined by the conditions under which the oil is to be used. The purpose of lubrication is to secure and maintain a film of oil sufficient to keep the bearing surfaces apart at working temperatures. If an oil used on a bearing is too light in viscosity at working temperature to sustain the load it will permit metal contact, with a consequent bearing wear and loss of power. On the other hand, if an oil which has too great body at working temperature is used, there will be unnecessary fluid friction, and again there will be loss of power. The more viscous the lubricant, the greater the bearing pressure which can be sustained; the viscosity of the lubricant should, therefore, be in proportion to the bearing pressure. Heavy, slow-moving machinery requires an oil of high viscosity; light, swiftly-moving machinery requires just the opposite. Also, high temperature conditions require an oil of higher viscosity than ordinary conditions. Most of the resistance or lost power in running machinery is caused by the internal friction or the fluid friction of the lubricant used. It is only occasionally, through the use of an improper oil or lack of feed of a proper oil, that actual metal wear on properly constructed bearings takes place.

The general law governing lubrication is that within certain limits a reduction in the viscosity of a lubricant will bring about a reduction in the coefficient of friction, or, in other words, a reduction in frictional heat and in lost power.

*Carbon.*—While petroleum or any petroleum product is a mixture of hydrocarbons, it must be remembered that the hydrocarbons themselves are chemical compounds. The car-

bon and hydrogen are combined chemically in different proportions to form entirely new and different substances called hydrocarbons. To say that an oil contains carbon means as much as to say that a piece of chalk contains carbon or that water contains hydrogen gas. To be sure, carbon is one of the chemical constituents, but it has absolutely lost its identity, and before the carbon can be reclaimed the compound must be decomposed or broken up into its constituent elements.

When any substance containing carbon is subjected to high temperature without a free supply of air, the substance may be decomposed. If so, the gaseous elements are liberated or combine with other elements to form new compounds, and a residue is left which is chiefly carbon. In most instances, however, this would not be pure carbon, but would consist of various other substances as well. Since this decomposition takes place to a greater or less degree when lubricating oils are subjected to certain conditions, the question of the decomposition of hydrocarbons and the deposit of carbon as a residue is an important one in the lubricating problem.

There is a popular idea that all deposits from a lubricating oil are due to carbonization. This, however, is not the case, as the deposits which are sometimes found in bearings or oiling systems are due to ordinary wear, and consist of small pieces of metal due to metallic wear or iron rust combined with dust and deposits from the surrounding air, and do not contain carbon from the oil unless the bearing has burned out. In the case of steam cylinder lubrication the deposits are due rather to boilerwater impurities and to small amounts of metal, mostly from rust, which are bound together by the oil. In course of time this oil partially evaporates and decomposes, leaving a hard deposit which contains only a small amount of oil and a relatively small percentage of carbon. An analysis of an average deposit from a steam cylinder might show that the residue contained 2 per cent of unchanged oil, 1 per cent of decomposed hydrocarbons, 7 per cent of carbonaceous matter

of carbon, and 90 per cent of ash, consisting of metallic substances, rust, dust, boiler impurities and dirt.

While the deposits found in an internal-combustion engine are by no means all carbon, still the percentage of carbon in the deposit is large, and it is in this case that the question of carbon deposits is of the greatest importance. Any deposit in the cylinder and on the top of the piston and on the valves causes a great amount of inconvenience and damage, and therefore special stress is laid on the amount and nature of the carbon deposited in use for internal-combustion work.

*Water and Sediment.*—The presence of water in lubricating oil is a matter that is often brought forcefully to the attention of those who handle petroleum products. The same may be said with respect to sediment. Not all of the impurities in petroleum products, however, are included under this heading. Others may be due to improper refining, or to improper or careless handling of the finished products, but their detection usually requires a more exhaustive test or analysis than is possible without the use of a laboratory.

Water in any petroleum product may have come from various sources. It may have been present when the oil was shipped by the manufacturer, it may have entered during transit, or it may have entered at the consumer's plant. It is almost invariably true that refined petroleum products are free from water when they leave the refinery. It is only the rarest exception that proves the rule. But when one considers the handling which the finished product necessarily receives, the piping, storing and transportation, also that it is often necessary, in the case of the heavier products, such as cylinder stocks, to keep the same in tanks where steam coils have been installed, it can be seen readily how easy it is for water to be introduced.

The chances of water being introduced into oil at the consumer's plant are greater than at the plants of the manufacturer. Whenever water is found in lubricating oil at the con-

sumer's plant, a careful examination of the conditions under which the oil is stored or used should be made in order to determine its source. A sample of at least one gallon of the water or mixture of water and oil should be obtained, and if any water from a pipe or faucet could have leaked into the storage tank or could have been in the oil can, a sample of this water should also be taken. The laboratory may then be able to determine whether the water in each case is from the same source. Samples of the oil from the original package should also be secured if possible. Special care should be taken, however, to prevent a sample being taken and placed in a bottle which is not perfectly clean and dry. In obtaining samples from the packages, a glass tube or thief should be used. This should be carefully cleaned and dried before taking samples from the different packages or different grades of oil, and should be cleaned after each sample is taken.

Many complaints concerning the presence of water in lubricating oils have been investigated and in nearly every case the water complained of was of purely local origin, and too much care cannot be taken by those interested in preventing water getting into petroleum products.

The question as to the source of sediment is not as easily answered as that concerning water. Sediment may consist of almost anything. When steel drums are used, a small amount of white sediment, due to the imperfect galvanization of the drums, will sometimes be found. In galvanizing the drums the interior surface is coated with molten zinc. If this process is not perfect the zinc coating will corrode, an action similar to the rusting of iron. Iron gives a red or brown rust, however, while zinc gives a white powder known as zinc white.

In the case of wooden barrels, the usual source of sediment is the glue which is used to coat the inside of the barrel in order to make it oiltight. When this glue has set it is very brittle, and if a heavy sledge is used to start the bung, there is always the possibility that fine particles of the glue lining

will fall into the oil. This is a trouble which the consumer may easily avoid by seeing that barrels of oil of any kind, especially lubricants, are not handled too roughly when being opened. Sometimes the bungs are driven in, and as the bung is made tapering, the outside end, being of larger diameter, often carries in chips and pieces of glue from around the inside of the bung hole. For this reason bungs should not be driven into the barrel, but the stave in which the bung is placed should be lightly tapped until the bung is loosened. Sometimes the barrel head is bored with an auger in order to insert the faucet. If this is not done in a careful manner chips may fall into the oil. In some instances, when oil is pumped from the bung hole, waste is used to wipe off the pump when it is removed, and any oil slopped around the bung hole is also wiped off in this manner. Small particles of waste are thus introduced into the barrel and subsequently a complaint may be made that the oil contained waste. Waste is never used in barrel-filling houses of manufacturers, but is used quite generally in the engine room, and therefore waste found in oil is evidence that the contamination occurred after the oil reached the plant of the consumer. All lubricating oils are carefully strained before being barreled and every precaution is taken to have them perfectly clean when shipped.

In considering water and sediment in lubricating oil it is necessary to distinguish clearly between impurities found in the unused oil and those taken up in use. In circulating or gravity-feed systems supplying oil to either horizontal or vertical engines, a certain amount of water leaks from the piston rods into the circulating oil. This leakage or condensation is, at times, very troublesome, especially where compounded cylinder oils are used, in which event the small amount of fixed oil causes the circulating oil to form an emulsion with the water. Experiments have also proven that the various boiler compounds used to soften the boiler water sometimes come over with the steam or water in priming or

because of foaming boilers. These compounds probably deposit in the cylinders and are afterwards washed out through the stuffing boxes into the circulating oil, having a very bad effect upon the oil, especially if it is mixed with water. Such conditions, of course, make it very difficult for the oil to remain in its original state and do its work properly. This can be remedied, however, by seeing that the stuffing boxes are tight, or by fitting up an arrangement to catch any leaking water so that it will not get into the circulating oil.

In a circulating system water or sediment accumulated in use should be removed by the filters. However, in order that these impurities may be removed from the circulating oil it is necessary to have a circulating system sufficiently large to allow time for the water and sediment to settle out. In many instances the reason for the breakdown of an oil in a system is the fact that it is constantly circulated at high speed with water and dirt, the trouble being due to the size and construction of the system and not to the oil.

Frequently deposits found in bearings are attributed to impurities in the unused oil. An analysis of such a deposit reveals the fact that it is due to impurities accumulated in use and that these impurities are entirely foreign to the oil as received from the manufacturer. These deposits consist of small pieces of metal and dirt which have been collected by the oil in use. The oil acts as a binding or collecting agent for all of the dirt around the plant, for impurities in the boiler water or waste, and the greatest care should be exercised at all times to prevent, to the greatest possible degree, the accumulation of water and sediment in lubricating oils.

*Emulsification Tests.*—Emulsion tests are made on all straight mineral oils, except cylinder oils. Four emulsion runs are made, usually 40 c.c. of oil in each case and

(1) 40 c.c. of distilled water;
(2) 40 c.c. of salt water;
(3) 40 c.c. of normal caustic soda;
(4) 40 c.c. of boiling distilled water.

The mixture is stirred with a paddle for five minutes at 1,500 revolutions per minute, the mixture being kept at a temperature of 130 degrees F. during the stirring and while settling out. On oils used with forced lubrication or on ice machines the oil must completely settle out in less than 30 minutes.

In concluding this treatise would say that the oil manufacturers are today handling the question of lubrication on the most scientific basis in the way of studying all new devices for the efficient handling of lubricants in operation, and by practical demonstrations this has been accomplished by means of a corps of skilled engineers who are trained carefully in the lubricating business in connection with the laboratory and refinery, using in such demonstrations a number of delicate instruments of all types of mechanism in actual service which proves conclusively the results of their investigations. The system of introducing and selling oils as has prevailed in the past is now becoming obsolete, and the scientific method is being employed. This can have no other result than aiding in the progress of mechanical efficiency, lessening worry and labor of the operating engineer, and giving them subjects to study and improve upon such as they have in general mechanical development where quality is the first consideration.

# ENGINES FOR CONVERSION OF ENERGY INTO USEFUL WORK.

By Commander J. O. Fisher, U. S. Navy, Member.

Bacon in his "Novum Organum" states "Man as a minister and interpreter of nature is limited in act and understanding by his observation of the order of nature. Neither his knowledge nor his power extends further." In this statement Bacon recognizes limits between which man must work in understanding the forces of nature and that his understanding is limited by his observations, which, in turn, are limited by his senses.

Man, being a rational animal, reasons and thinks, but his reason and his thought are applied to facts which he observes through his senses.

The engineer must recognize that the acid test for any fact is "Can this fact be established by experiment to our senses?" a demonstration which nature never omits in the trial of the completed machine.

Man observes the universe as being space filled with matter, and since it is matter or the qualities of matter which affect man's senses, the absence of matter cannot be established to his senses (*i. e.*, that while man may assume that space can exist without matter, this is an assumption and not a fact). Man has in the past assumed that a perfect vacuum was possible after extracting all matter from space, only to find that with improved apparatus and by other aids which have been developed, it has later been established that the space still contained matter in minute quantity. Assumptions may be desirable for some particular reason, but when reaching a con-

clusion which involves their use it should never be forgotten that such an assumption was made and affects the conclusion reached.

Exactly as man cannot establish to his senses the existence of space without matter neither can he establish to his senses the existence of matter without space, and since to his senses all space is fixed and invariable he accepts the Law of Conservation of Matter that "the total quantity of matter in the universe is fixed and constant and matter cannot be created or destroyed" as being a rational conclusion based on his observation.

In the study of physical science man has found it necessary and desirable to measure portions of matter by an operation which he calls weighing, and weight is a measure of the relative force with which any portion of matter at any position is attracted by that portion of matter which we call the earth, as compared to the force acting on a specified volume of specified matter at a particular relative position.

The laws having to do with this attraction of one portion of matter for another were discovered and enunciated by Isaac Newton, and are as follows:

1. Every body perseveres in its state of rest or of moving uniformly in a straight line except in so far as it is made to change that state by external forces.

2. Change of motion is proportional to impressed forces and takes place in the direction in which the forces act.

3. Reaction is always equal and opposite to action—that is to say, the action of two bodies upon each other is always equal and in opposite directions.

The law of the conservation of matter and Newton's laws are inseparably connected and it is through the recognition of this conservation law that most new elements of matter have been discovered. That is, if a certain quantity of matter undergoes several changes in structure the quantity of matter remaining should equal the quantity of original matter used

in the operations. After several operations and identification of different portions of matter in their final form as belonging to certain known substances, matter has been found to be missing or present in excess, and in identifying this missing or excess quantity the new elements have been discovered.

A little consideration and thought given to the above shows that while we consider the law of the conservation of matter as established we apply this law by measuring relative forces, and in reality we distinguish between the different elements of matter by their specific gravity, which is a measure of the force of gravity per unit of space occupied by particular matter, and it is rational to regard the universe as made up of different concentrations of force, or, as we say at the present time, of different kinds of matter.

Faraday in discussing this wrote a long dissertion which he delivered before the Philosophical Society called "The Conservation of Force," in which he showed that the conservation of force and the conservation of matter were one and the same (*i. e.*, force could not be separated from matter and matter could not be separated from force), and if the law of the conservation of matter was true it must follow that the law of conservation of force is true.

Faraday also delivered lectures on "The Forces of Nature" in which he demonstrated by experiment how forces, although differing in their characteristics (*i. e.*, effect on our senses), could be converted one into the other. The forces enumerated by him were gravity, gravitation, cohesion, chemical affinity, heat, magnetism and electricity; and the thought arises that the different characteristics of these forces may rest in our means of observing them and is not a real difference (*i. e.*, we observe different forces with different senses, and as they are recognized by their effect on one sense or the other we give them different names).

It is conceivable that there is an ultimate element of matter of which what we call elements at the present time are com-

posed, and the delay in the discovery of this ultimate element is due to the handicap of our senses, and it is also conceivable that the different forces of nature are really combinations or derivations of one ultimate force, which in different combinations is recognized as a different force by our senses.

Prior to the development of the science of chemistry and to the recognition of the law of conservation of matter, new discoveries were haphazard and their establishment on the firm foundation of experimental fact was doubtful. With the development of standards for determining their characteristics and the recognition of the law of conservation of matter new discoveries established on a firm foundation of experimental fact have continued to make their appearance.

Faraday in summing up in his lecture on "The Conservation of Force" states as follows:

"Just as the chemist owes all the perfection of his science to his dependence on the certainty of gravitation applied by the balance, so may the physical philosopher expect to find the greatest security and the utmost aid in the principle of the conservation of force. All that we have that is good and safe, as the steam-engine, the electric telegraph, etc., witness to that principle. It would require a perpetual motion, a fire without heat, heat without a source, action without reaction, cause without effect, or effect without a cause, to displace it from its rank as a law of nature."

Joule in establishing the mechanical equivalent of heat energy, and other discoverers and experimenters who have established units for conversion of energy measurements, have proceeded along the same lines as regards force in combination with matter, and all their discoveries are based upon the Law of the Conservation of Energy, that "the quantity of energy in the universe is constant and energy cannot be created or destroyed."

Therefore since

(*a*) Man cannot establish to his senses the existence of space without matter, or the existence of matter without space.

(*b*) Neither can man establish to his senses the existence of matter without force, or the existence of force without matter and,

In the same manner of observation of nature, man observes action and reaction, cause and effect, motion and position, and in all cases his observations cover an interval of time. Man can assume the interval of time to be infinitely small, but the elimination of that interval is impossible. Therefore we may conclude that—

(*c*) Man cannot establish to his senses the existence of action without time or the existence of time without action.

From (*a*) is derived the LAW OF CONSERVATION OF MATTER or space that "THE QUANTITY OF MATTER OR SPACE IN THE UNIVERSE IS CONSTANT AND IT CANNOT BE CREATED OR DESTROYED."

From (*b*) is derived the LAW OF THE CONSERVATION OF ENERGY (space plus force) that "THE QUANTITY OF ENERGY IN THE UNIVERSE IS CONSTANT AND IT CANNOT BE CREATED OR DESTROYED."

From (*c*) is derived the LAW OF THE CONSERVATION OF POWER (space plus force plus time) that "THE QUANTITY OF POWER IN THE UNIVERSE IS CONSTANT AND IT CANNOT BE CREATED OR DESTROYED."

These CONSERVATION LAWS constitute the basis for all investigations of the forces of nature and their use for the benefit of mankind and on them is erected the entire structure of Engineering.

Therefore to our observation the universe consists of space, force and time, and from the arbitrary limits imposed by our senses, we have derived and accept the CONSERVATION LAWS enumerated above.

To assist in rational discussion definition of terms is necessary and the following are submitted:

*Matter* may be defined as sensible space.

*Force* may be defined as a tendency to change sensible space (*i. e.*, matter), its motion, position, volume, structure and force as contained in relative volumes.

*Energy* is space plus force.

*Power* is space plus force plus time.

Fundamentally we have only three measurements in the universe: *Force, Space* and *Time.*

In Engineering we have—

Force.

Energy or work (whose factors are force and space).

Power (whose factors are force, space and time).

In the study of Engineering and the conversion of energy into useful work by means of an engine, it is desirable that we consider energy and not force. If a chemist were to consider only one element and neglect the different combinations with the other elements, he would naturally get into error. In the same way an engineer in considering only one of the forces acting to produce certain results and neglecting the others, is sure to get into error.

An instance of this in the study of heat engines and the science of thermodynamics is Carnot's heat engine from which he derives the "Carnot Principle" which was written many years ago.

The following description of Carnot's heat engine and its method of operation is taken from the "Encyclopedia Britannica" and the notes and comment are made in an endeavor to point out the errors in reasoning and fact and the neglect of forces other than heat which make the conclusions reached of doubtful value.

CARNOT'S HEAT ENGINE.

I. Carnot: On the Motive Power of Heat.—A practical and theoretical question of the greatest importance was first answered by Sadi Carnot about this time in his "Reflections

on the Motive Power of Heat" (1824). How much motive power (defined by Carnot as weight lifted through a certain height) can be obtained from heat alone by means of an engine repeating a regular succession or "cycle" of operations continuously? Is the efficiency limited, and, if so, how is it limited? Are other agents preferable to steam for developing motive power from heat?

II. Carnot points out that in order to obtain an answer to this question, it is necessary to consider the essential conditions of the process, apart from the mechanism of the engine and the working substance or agent employed. Work cannot be said to be produced *from heat alone* unless nothing but heat is supplied, and the working substance and all parts of the engine are at the end of the process in precisely the same state as at the beginning.

Note.—To make this paragraph more definite it might be added "And in equilibrium as regards temperature, pressure, etc., and at rest,—(*i. e.*, no unbalanced forces)."

III. Carnot's Axiom.—Carnot here, and throughout his reasoning, makes a fundamental assumption, which he states as follows: "When a body has undergone any changes and after a certain number of transformations is brought back identically to its original state, considered relatively to density, temperature and mode of aggregation, it must contain the same quantity of heat as it contained originally."

Note.—Provided this axiom is correct it must also be true that "when a body has undergone any changes in state, considered relatively to density and mode of aggregation at a constant temperature, it must contain a quantity of heat differing from the quantity of heat it contained originally."

IV. Heat, according to Carnot, in the type of engine we are considering, can evidently be a cause of motive power only by virtue of changes of volume or form produced by alternate heating and cooling. This involves the existence of cold and hot bodies to act as boiler and condenser, or source and sink

of heat, respectively. Wherever there exists a difference of temperature it is possible to have the production of motive power from heat; and conversely, production of motive power, from heat alone, is impossible without difference of temperature. In other words the production of motive power from heat is not merely a question of the consumption of heat, but always requires transference of heat from hot to cold. What, then, are the conditions which enable the difference of temperature to be most advantageously employed in the production of motive power, and how much motive power can be obtained with a given difference of temperature from a given quantity of heat?

Note.—Changes of volume accompany changes in heat content, but all changes of volume are not caused by heating and cooling. In some cases, for instance, the heating and cooling are the result of changes in volume. The necessity for the existence of cold and hot bodies is not clear. If we assume compressed air at a pressure greater than atmosphere and a temperature equal to T, that volume of air will expand (increase in volume) against the pressure of the atmosphere, doing work, and its temperature will fall. If kept in contact with a source of heat at the temperature T its temperature will remain constant, and in this particular case we have a change of volume with no heating and no cooling, and a transfer of heat energy to the working substance which does work by virtue of its original internal energy, utilizing the heat received in maintaining its temperature. This change in volume is accompanied by a corresponding conversion of energy into work, the energy supplied by operating substance being made up by heat from the source of heat and the flow from the source to the operating substance being caused by the difference in temperature. This phenomena is one that can be verified by experiment, and it seems to nullify nearly every statement in the preceding paragraph, being at constant temperature, except that the compressed air during expansion and

while in contact with the source of heat will be at a lower temperature than the source of heat, but this difference in temperature can be assumed as infinitely small.

V. Carnot's Rule for Maximum Effect.—In order to realize the maximum effect, it is necessary that, in the process employed, there should not be any direct interchange of heat between bodies at different temperatures. Direct transference of heat by conduction or radiation between bodies at difference temperatures is equivalent to wasting a difference of temperature which might have been utilized to produce motive power. The working substance must throughout every stage of the process be in equilibrium with itself (*i. e.*, at uniform temperature and pressure) and also with external bodies, such as the boiler and condenser, at such times as it is put in communication with them. In the actual engine there is always some interchange of heat between the steam and the cylinder, and some loss of heat to external bodies. There may also be some difference of temperature between the boiler steam and the cylinder on admission, or between the waste steam and the condenser at release. These differences represent losses of efficiency which may be reduced indefinitely, at least in imagination, by suitable means, and designers had even at that date been very successful in reducing them. All such losses are supposed to be absent in deducing the ideal limit of efficiency, beyond which it would be impossible to go.

Note.—Why is direct transference of heat by conduction or radiation between bodies at different temperatures equivalent to wasting a difference of temperature which might have been utilized to produce motive power? It has not been demonstrated that temperature or difference in temperature produces motive power. Temperature is a measure of relative intensity of heat force and indicates the relative intensity of this force in different matter, and as such indicates the possibility or impossibility of the transfer of heat. In the preceding paragraph the statement is made that production of motive

power from heat alone is impossible without difference in temperature * * * and always requires transference of heat from hot to cold. In this paragraph it is stated that in order to realize the maximum effect there should not be any direct interchange of heat between bodies at different temperatures. This is a most ridiculous form of reasoning. First, the fact is stated (*i. e.*, that a difference in temperature is necessary to transfer heat), and next a goal (*i. e.*, transfer of heat without difference of temperature to prevent waste) is established, which involves the destruction of the previous fact. It is certainly "reductio ad absurdum" to establish a fact that heat cannot be transferred without a difference in temperature and then to indicate the possibility of developing a more efficient heat engine provided this demonstrable fact is eliminated. All of which goes hand in hand with the assumption that temperature can be utilized to produce motive power which under ordinary conception of temperature is as impossible, as it is for velocity or pressure or position to produce motive power. The assumption and statement that a difference of temperature represents a loss of efficiency is the answer to the question we are investigating and trying to establish by reasoning on fact.

VI. Carnot's Description of His Ideal Cycle.—Carnot first gives a rough illustration of an incomplete cycle, using steam much in the same way as it is employed in an ordinary steam engine. After expansion down to condenser pressure the steam is completely condensed to water, and is then returned as cold water to the hot boiler. He points out that the last step does not conform exactly to the condition he laid down, because, although the water is restored to its initial state, there is direct passage of heat from a hot body to a cold body in the last process. He points out that this difficulty might be overcome by supposing the difference of temperature small, and by employing a series of engines, each working through a small range, to cover a finite interval of temperature. Having established the general notions of a perfect cycle, he proceeds to

give a more exact illustration, employing a gas as the working substance. He takes as the basis of his demonstration the well-established experimental fact that a gas is heated by rapid compression and cooled by rapid expansion, and that if compressed or expanded slowly in contact with conducting bodies, the gas will give out heat in compression or absorb heat in expansion while its temperature remains constant. He then goes on to say—(par. VII below)

Note.—In his rough illustration of an incomplete cycle using steam, Carnot fails to show how any reduction in pressure of the steam in the cylinder takes place or how expansion can take place except against a lower pressure in condenser, both questions being neglected. Having a constant pressure on his piston, the expansion of steam at constant temperature against a constant pressure does work at constant pressure and constant temperature. After reaching a point, however, at which all water is steam at atmospheric pressure, further expansion cannot take place unless the pressures on opposite sides of the piston are unbalanced. If the pressure outside is decreased further expansion is possible, but that in itself involves an expenditure of power. If the pressure inside the cylinder (*i. e.*, operating substance) is decreased slightly the pressure on the outside of the piston would operate to compress the operating substance, and this compression can take place at constant temperature and constant pressure, approximately at exactly the same pressure and temperature at which the expansion took place. In this case no useful work is done, and this is what Carnot has established as the general notion of a perfect cycle, admitting that when the steam passes into the condenser, this step having to do with the transfer of heat, does not conform to his requirements as regards the heat cycle. He neglects all mention of pressure which is the operating force generated by the heat change and which, by establishing unbalanced forces acting on the piston, is the indirect means by which the heat transferred to the water, converting it into steam, does work.

In the last statement in this paragraph the fact is omitted that the expansion and contraction of the gas takes place by virtue of the pressure which it exerts or a pressure which is exerted on it, the origin of which is neglected. Heat given out in compression is generated by the work of compression which is obtained from some external store of work, and the heat which is absorbed by the gas in expansion replaces internal energy contained originally by the gas which has been converted into work during expansion. To make an incomplete statement of facts observed in any phenomena and to draw general conclusions therefrom is most dangerous to any investigation of physical truths. For instance, we know that air is supplied to a furnace for the generation of heat, but to draw the inference from this that heat is produced by supplying air to a furnace is so obviously an incomplete statement of fact that few people would make this error. The introduction of air to the furnace is an accompanying circumstance to the introduction of coal and fire, and both are as necessary as the air in generating heat. And we know that a gas will give out heat in compression and absorb heat in expansion while its temperature remains constant, but it can only be compressed by work expended in compressing it and the giving out of heat in compression while its temperature remains constant is caused by the work expended in compressing it. The loss of heat in compression at constant temperature is a result or effect and not a cause. Of course a decrease in volume or compression could be caused by a loss of temperature, which is also loss of heat, but that a gas will give out heat in compression or absorb heat in expansion at constant temperature implies work received from or delivered to some store of work. No transfer of heat at constant temperature during compression or expansion can take place through a transfer of heat only, and work expended or supplied should be mentioned in any statement of facts having to do with this operation.

In paragraph VI Carnot's illustration of an incomplete cycle

uses steam as the operating substance. In the latter part of the same paragraph he refers to a cycle using gas as the working substance. In paragraph VII he uses atmospheric air enclosed in a cylinder as the operating substance. The analogy between the steam engine and Carnot's original statement (see paragraph II) that heat only shall be supplied is not definite, and there is this difference between his statement of conditions in paragraph VII and his illustration of the cycle using steam in paragraph VI. In a steam engine the steam is added at a certain temperature and pressure to the cylinder, the mass of steam is increased during the addition of energy, whereas in the air engine the mass of the air in the cylinder, as per paragraph VII, is constant and the specific heat (*i. e.*, heat per unit mass) increases, otherwise there would be no additional heat. In other words, in a steam engine the mass of the operating substance is increased during the period that heat (*i. e.*, heat energy) is being added, and it is perfectly clear that the "heat" which Carnot refers to in his steam engine and in his air engine, per paragraph VII, are not one and the same. In one case the heat and its carrier, the operating substance, are added to the cylinder contents at the same time. In the other case the heat only is separated from its source and is added to the operating substance.

In the steam engine heat may be considered as energy, as it consists not only of heat but mass. In the air engine heat is a quality of matter, which is subtracted from the source A and added to the operating substance. In this case heat is force, and its transfer is a transfer of force and not energy. In the steam engine the heat is space plus force, in the second case it is force only.

By transfer of mass as in a steam engine work can be done at variable mass, at constant temperature and constant pressure which is at CONSTANT SPECIFIC HEAT (*i. e.*, heat per unit mass), whereas in the air engine work can be done at constant mass, constant temperature and variable pressure which

is at VARIABLE SPECIFIC HEAT (*i. e.*, heat per unit mass). Attention is invited to the fact that in the following illustration of the operation of these engines Carnot uses heat: (1) as being an intangible quality of matter which can be transferred from the source A to the operating substance, and: (2) that this intangible quality of matter is transferred from the source to the operating substance and is the transfer of an intangible quality of matter and not a transfer of operating substance, (*i. e.*, a transfer of force and not energy).

VII. "This preliminary notion being settled, let us imagine an elastic fluid, atmospheric air for example, enclosed in a cylinder *abcd*, Fig. 1, fitted with a movable diaphragm or piston *cd*. Let there also be two bodies A, B, each maintained at a constant temperature, that of A being more elevated than that of B. Let us now suppose the following series of operations to be performed:

Note.—In this paragraph of assumptions regarding the engine no mention is made of the piston being in equilibrium. So far as the statement itself goes the piston can be moving or in equilibrium, or at rest held in position against some force or pressure of the air in the cylinder (*i. e.*, in his description of the operation of the engine the detailed conditions specified are not complete).

"1. Contact of the body A with the air contained in the space *abcd*, or with the bottom of the cylinder, which we will suppose to transmit heat easily. The air is now at the temperature of the body A, and *cd* is the actual position of the piston.

"2. The piston is gradually raised, and takes the position *ef*. The air remains in contact with the body A, and is thereby maintained at a constant temperature during the expansion. The body A furnished the heat necessary to maintain the constancy of temperature."

Note.—What raises the piston? Is it the expansion of the air? and if so, how can the air expand when it is at constant

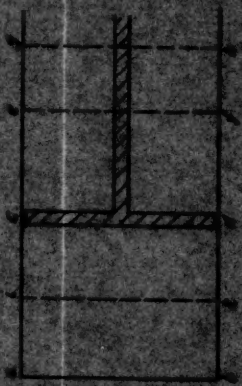



FIGURE 1

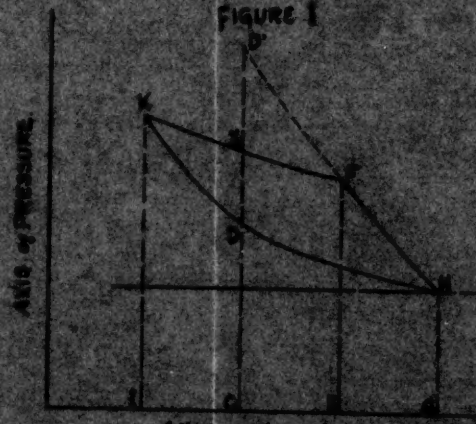
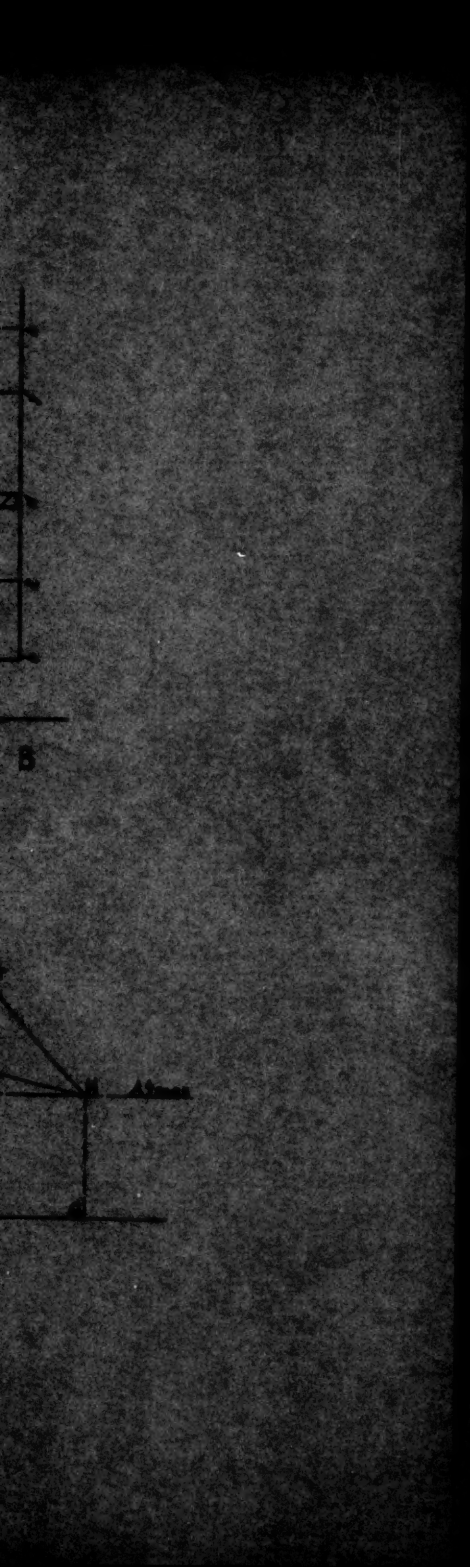



FIGURE 2.


B

temperature, except by virtue of pressure? That is, the expansion takes place not by virtue of the heat received from the source, but by virtue of the pressure of the air, and as the air expands its temperature decreases and absorbs heat from the source A, and in that way the air is maintained at a constant temperature during the expansion.

"3. The body A is removed, and the air no longer being in contact with any body capable of giving it heat, the piston continues nevertheless to rise, and passes from the position *ef* to *gh*. The air expands without receiving heat and its temperature falls. Let us imagine that it falls until it is just equal to that of the body B. At this moment the piston is stopped and occupies the position *gh*."

Note.—What makes the piston continue to rise except pressure, and the pressure, as we have seen, was not derived from the source of heat but was contained in the air at starting. Until the body A was removed the temperature of the air had not changed, and consequently it could not have had its pressure increased due to an increase in temperature. It could and probably did, however, have its pressure decreased by an increase of volume at constant temperature. The air expands, but it can only expand when the pressure on opposite sides of the piston is unbalanced. However, after expanding to a point at which its temperature is just equal to the body B we can assume that the pressure on opposite sides of the piston is equal.

"4. The air is placed in contact with the body B; it is compressed by the return of the piston, which is brought from the position *gh* to the position *cd*. The air remains meanwhile at a constant temperature, because of its contact with the body B to which it gives up its heat."

Note.—What returns the piston? The air will lose no heat to the body B and the piston will not move except under the action of force, and any heat that is transferred to the body B must come from some source. It can't be created by the air—

it can't be created by the engine—it can only be created by the expenditure of mechanical energy in compressing the air through pressure exerted on the piston. In other words, during this operation the original limitation that heat alone shall be supplied to the engine has been violated unless it is assumed that the work used in compression is derived from the work done in expansion, in which case a liberal interpretation would concede that heat only had been received from an outside source and part of the work done by this heat during expansion had been used during compression in the cycle of operations. To receive heat from an external source at a high temperature to do work and then to use some of that work to deliver heat to a sink at a lower temperature does not appear to me an efficient operation, but rather as progressing backward, in that an engine is used to transfer heat from a source at a high temperature to a sink at a lower temperature when by placing the two in contact it could be transferred much quicker, and more efficiently and without using an engine. What is the particular advantage derived from the operation?

"5. The body B is removed, and the compression of the air is continued. The air being now isolated, rises in temperature. The compression is continued until the air has acquired the temperature of the body A. The piston passes meanwhile from the position *cd* to the position *ik*."

Note.—The compression of the air in this operation involves a further drain on our store of work which we obtained during expansion. This differs from sub-paragraph 4 in that all the work is delivered to the air and is not delivered as heat to some outside sink.

"6. The air is replaced in contact with the body A, and the piston returns from the position *ik* to the position *ef*, the temperature remaining invariable.

"7. The period described under (3) is repeated, then successively the periods (4), (5), (6); (3), (4), (5), (6); (3), (4), (5), (6); and so on.

"During these operations the air enclosed in the cylinder exerts an effort more or less great on the piston. The pressure of the air varies both on account of changes of volume and on account of changes of temperature; but it should be observed that for equal volumes, that is to say, for like positions of the piston, the temperature is higher during the dilation than the compression. Since the pressure is greater during the expansion, the quantity of motive power produced by the dilatation is greater than that consumed by the compression. We shall thus obtain a balance of motive power, which may be employed for any purpose. The air has served as working substance in a heat engine; it has also been employed in the most advantageous manner possible, since no useless re-establishment of the equilibrium of heat has been allowed to occur."

Note.—Statements in this paragraph seem plausible, but the statement that the temperature is higher during the dilation than the compression is one which rests upon the assumption that expansion or dilation at constant temperature through the addition of heat only is a possible physical operation. While as an assumption there are no objections to this, to draw a conclusion that a result is true which depends upon the truth of this assumption is not justified. The same comment applies to the statement "Since the pressure is greater during the expansion, etc." There is no experiment which has ever been made which will establish the fact that the pressure during the expansion of a given volume and mass of gas between certain temperature and volume limits is greater than the pressure during compression of the same volume and mass of gas between the same temperature and volume limits, provided that after expansion and compression the body is brought back identically to its original state, considered relative to density, temperature and mode of aggregation. The statement that "a balance of motive power has been obtained" may or may not be true. It has certainly not been demonstrated. The further statement that no useless re-establishment of the equilibrium

of heat has been allowed to work is not clear. In the process heat has been transferred from a source at high temperature to a sink at a lower temperature. Whether or not this is employing heat "in the most advantageous manner possible" in a heat engine is, to say the least, doubtful.

VIII. "All the operations above described may be executed in the reverse order and direction. Let us imagine that after the sixth period, that is to say, when the piston has reached the position *ef*, we make it return to the position *ik*, and that at the same time we keep the air in contact with the hot body A; the heat furnished by this body during the sixth period will return to its source, that is, to the body A, and everything will be as it was at the end of the fifth period. If now we remove the body A, and if we make the piston move from *ik* to *cd*, the temperature of the air will decrease by just as many degrees as it increased during the fifth period, and will become that of the body B. We can evidently continue in this way a series of operations the exact reverse of those which were previously described, it suffices to place oneself in the same circumstances and to execute for each period a movement of expansion in place of a movement of compression, and vice versa.

IX. "The result of the first series of operations was the production of a certain quantity of motive power, and the transport of heat from the body A to the body B; the result of the reverse operations is the consumption of the motive power produced in the first case, and the return of heat from the body B to the body A, in such sort that these two series of operations annul and neutralize each other."

Note.—That the operations above described can be executed in the reverse order and direction means nothing except that energy cannot be created or destroyed. To state that a process for operating an engine is reversible is merely a statement that energy cannot be created or destroyed. The above paragraph assumes, however, that with the operating substance at the

lower temperature it is possible to return it to the higher temperature. And since the engine receives heat only, we have a plausible statement that heat can be transferred from a cold body to a hot one by a purely self-acting process. The engine may be reversible, but only when it receives work from an external store of work, which violates the original premise in that heat only is received.

X. "The impossibility of producing by the agency of heat alone a quantity of motive power greater than that which we have obtained in our first series of operations is now easy to prove. It is demonstrated by reasoning exactly similar to that which we have already given. The reasoning will have in this case a greater degree of exactitude; the air of which we made use to develop the motive power is brought back at the end of each cycle of operations precisely to its initial state, whereas this was not quite exactly the case for the vapor of water, as we have already remarked."

Note.—The statements in this paragraph are ridiculous. Having performed a series of operations with an engine which violates the original premise and the operation of which is based upon assumptions, it is now stated that it is impossible to produce by the agency of heat alone, a quantity of motive power greater than that which we had obtained in our first series of operations. Having obtained a perfect cycle of operations by certain assumptions, it is impossible to assume anything that will improve the original assumption, and consequently the conclusion reached is that a perfect cycle of operations is performed. This is ridiculous.

XI. "Carnot's Statement of His Principle.—If the above reasoning be admitted, we must conclude with Carnot that the motive power obtainable from heat is independent of the agents employed to realize it. The efficiency is fixed solely by the temperatures of the bodies between which, in the last resort, the transfer of heat is effected. We must understand here that each of the methods of developing motive power attains

the perfection of which it is susceptible. This condition is fulfilled if, according to our rule, there is produced in the body no change of temperature that is not due to change of volume, or in other words, if there is no direct interchange of heat between bodies of sensibly different temperatures."

Note.—Reason can be applied with surety only to fact. To reason upon assumptions is to fall into error. What is the efficiency? Is it the ratio of the net power delivered to the total heat supplied to the engine, or is it the ratio of the net work done to the net heat supplied? These points are not clear. The net heat supplied must be equal to the net work done, otherwise the Law for the Conservation of Energy has been violated. If the efficiency is the ratio of the total heat supplied to the total work, how is the work used in compression, which is negative work, accounted for? If efficiency is equal to the ratio of the net work delivered to the total energy supplied, why is the efficiency fixed solely by the temperatures of the bodies between which is the last resort the transfer of heat is effected? What reasoning on fact has established this principle? That the total quantity of heat which can be utilized by any operating substance in any engine is measured by the temperature limits is not true in fact, except the substance be at constant pressure. Its power of doing work is dependent more upon pressure limits than upon temperature limits and its pressure can co-exist with a low temperature as well as with a high temperature.

XII. "It is characteristic of a state of frictionless mechanical equilibrium that an indefinitely small difference of pressure suffices to upset the equilibrium and reverse the motion. Similarly in thermal equilibrium between bodies at the same temperature, an indefinitely small difference of temperature suffices to reverse the transfer of heat. Carnot's rule is therefore the criterion of the reversibility of a cycle of operations as regards transfer of heat. It is assumed that the ideal engine is mechanically reversible, that there is not, for instance, any

communication between reservoirs of gas or vapor at sensibly different pressures, and that there is no waste of power in friction. If there is equilibrium both mechanical and thermal at every stage of the cycle, the ideal engine will be perfectly reversible. That is to say, all its operations will be exactly reversed as regards transfer of heat and work, when the operations are performed in the reverse order and direction. On this understanding Carnot's principle may be put in a different way, which is often adopted, but is really only the same thing put in different words: The efficiency of a perfectly reversible engine is the maximum possible, and is a function solely of the limits of temperature between which it works. This result depends essentially on the existence of a state of thermal equilibrium defined by equality of temperature, and independent, in the majority of cases, of the state of a body in other respects. In order to apply the principle to the calculation and prediction of results, it is sufficient to determine the manner in which the efficiency depends on the temperature for one particular case, since the efficiency must be the same for all reversible engines."

Note.—There has been no process of reasoning or statement of fact which can be established by experiment which shows that the efficiency of a perfectly reversible engine is greater than the efficiency of any other engine. If a theoretical heat engine is not perfectly reversible the Law of Conservation of Energy is not followed in its operation, and to assume that the Law of the Conservation of Energy is a function solely of the limits of temperature between which an engine works is such a bald misstatement of experimental fact that it is hardly worthy of serious consideration.

To summarize the comment as made in the notes on the Carnot engine and principle as taken from the Encyclopedia Britannica, there are three points in particular which are contrary to the operations of nature and which lead into the manifest errors and conclusions derived from study of the Carnot engine and principle.

The first of these errors is an error in reasoning in that a co-existing phenomena is mistaken for a cause, for instance, expansion at constant temperature doing work. The statement is made that expansion at constant temperature doing work is possible provided the operating substance is in contact with a source of heat. It is not definitely stated that the addition of heat caused the expansion, neither is it definitely stated that the operating substance has an internal energy available as pressure to act on the piston. By omitting these statements of fact the inference is incorrectly drawn that the supply of heat at the same temperature as the operating substance can be used to do work. This is an untenable statement of fact. The expansion of the operating substance under its own internal energy moves the piston, does work, and results in a decrease in the temperature of the operating substance. After this decrease has become an appreciable amount heat flows from the source to the operating substance, maintaining its temperature. The heat does not do the work, it merely maintains the temperature of the operating substance replacing with heat the internal energy used. The work is done by virtue of the internal energy of the air, and if you assume the air at such a condition as regards its internal energy that the pressure on opposite sides of the piston is equal there will be no movement of the piston. This being so, how can these experimental facts be reconciled with Carnot's own statement that heat alone shall be supplied to the engine? Is internal energy present as pressure or heat, or is it energy which can be converted into heat?

The second point is the reverse of the first in that it is assumed that compression of the air can take place as a result of the transfer of heat only at constant temperature. The statement is made that air can be compressed at constant temperature provided it is in contact with a sink of heat at that temperature, the sink of heat receiving heat from the air under compression. It is not stated that the subtraction of heat

causes the compression, neither is it stated that compression takes place as a result of external work being done on the piston which, after causing an appreciable increase in the temperature of the operating substance, will cause a flow of heat to the sink at constant temperature in contact with the operating substance. The fact that there is a flow of heat at approximately constant temperature from the operating substance to the sink is interpreted as being the cause of action. A second time the same error is made and a coexisting phenomena is assumed as the cause of compression. Can this external work done on the piston be considered as heat? Does not Carnot violate his condition that heat alone shall be supplied in receiving external work on the piston?

The third point which is in error is the assumption that the efficiency of a reversible engine is greater than the efficiency of any engine not reversible. It is assumed apparently that there are heat engines which are not reversible, which is equivalent to a statement that the energy supplied to the engine and the energy derived from the engine are not equal (*i. e.*, heat and mechanical energy are not mutually convertible). A reversible engine is nothing more or less than an engine which operates in accordance with the Law of the Conservation of Energy. To assume that any engine can operate except in accordance with that Law is ridiculous, but hardly more so than to assume that because an engine does operate in accordance with that Law its efficiency is unity and a function solely of the limits of temperature between which it works.

When plotted on a pressure volume diagram the indicator card of Carnot's heat engine consists of two isothermal and two adiabatics. Referring to figure 2, lines KF and DH are isothermal lines and the lines FH and KD are adiabatic lines. Contrary to the usual diagram illustrating the Carnot cycle the atmospheric line is also added as indicating the pressure of the operating substance at maximum volume. The cycle can be examined mathematically provided (1) we accept the Law of

Conservation of Energy, and (2) we eliminate the inferences and make our assumptions in accordance with experimental fact. For instance, we can assume that when the piston in figure 1 is at the position *ik* the temperature of the operating substance shall be equal to the temperature of the source of heat A, and at a pressure indicated by the location of a point K on figure 2. We can then go through a complete cycle of operations as follows:

First operation.—Piston at position *ik*, operating substance at Temperature A and at pressure K. Source of heat A is placed in contact with the cylinder. The operating substance expands, driving the piston from the position *ik* to the position *ef* at which point the source of heat A is removed and the cylinder thermally insulated. By the Law of the Conservation of Energy the heat supplied must be equal to the work done by the piston plus the energy change in the operating substance. The work done equals area IKFE (see note), the heat received from A equals QA, the change in the energy content of the operating substance equals QF — QK. We have, in accordance with the Law of Conservation of Energy, our first equation,

$$(1)\quad QA = W\,IKFE + QF - QK.$$

Note.—The pressure during expansion of operating substance in driving piston from *ik* to *ef* is indicated by line KF on figure 2. The distance piston has moved is equal to the distance IE on volume line of P. V. diagram, figure 2. Since force times distance equals work or energy the work done is equal to the area IKFE and may be written W IKFE.

Second operation.—The cylinder is now thermally insulated but the pressure of the operating substance on the piston being greater than the pressure of the atmosphere the operating substance will expand doing work and driving the piston from the position *ef* to the position *gh* at which position the pressure of the operating substance is equal to the pressure of the atmosphere and the piston is in equilibrium and at rest. Work done

is equal to area EFHG and energy lost by the operating substance is equal to QF — QH, and since no energy has been received from any other source we have for our second equation in accordance with the Law of the Conservation of Energy:

(2) QF — QH = W EFHG.

Third operation.—The piston is at position *gh*, the operating substance is at temperature of the sink of heat B, and the pressure of the operating substance is equal to the pressure of the atmosphere. The sink of heat B is placed in contact with the operating substance and work which has been done during expansion is returned to the piston, driving the piston from position *gh* to *cd*, compressing the air, which remains at constant temperature, rejecting heat to the sink of heat B, until its condition is as shown by position D, figure 2, which corresponds to the piston position *cd*. During compression heat has been rejected to B equal to QB. The energy of the operating substance has been changed from QH to QD.

If we assume that during compression of operating substance, by movement of piston from *gh* to *cd*, the sink of heat B was not placed in contact, but that compression was adiabatic (*i. e.*, with operating substance and engine thermally insulated as during the second operation) the compression line would follow the adiabatic line HF to D′ and the work of compression would equal area GHD′C. Change in energy content QD′ — QH. If then the sink of heat B was brought in contact with operating substance heat would be rejected at constant volume condition until the condition of operating substance would be indicated at D, and we would have heat rejected QB equal to QD′ — QD in accordance with Carnot's Axiom.

In accordance with the Law of Conservation of Energy we have

(3) $W\overline{GHD'C} = QD' - QH.$

(4) $QB = QD' - QD.$

(5) $\overline{WGHD'C} - \overline{WGHDC} = \overline{WDD'H}.$

Fourth operation.—The sink of heat B is removed from contact with the operating substance. More work is borrowed from that done during expansion, and the piston is moved from the position *cd*, figure 1, corresponding to D, figure 2, to the position *ik*, figure 1, corresponding to the position K, figure 2, and during this operation the operating substance is thermally insulated and all work expended in compression is added to the operating substance as energy, the operating substance being returned to its original condition as indicated by K. During this operation work has been done equal to area CDKI. Energy has been added to the operating substance equal to QK — QD, and in accordance with the Law of the Conservation of Energy, we have:

(6) $QK - QD = W\overline{CDKI}.$

From these four operations we have derived the following equations:

(1) $QA = W\overline{IKFE} + QF - QK.$

(2) $QF - QH = W\overline{EFHG}.$

(3) $W\overline{GHD'C} = QD' - QH.$

(4) $QB = QD' - QD.$

(5) $\overline{WGHD'C} - \overline{WGHDC} = \overline{WDD'H}.$

(6) $QK - QD = W\overline{CDKI}.$

(7) Net heat received $= QA - QB.$

(8) Net work delivered $= W\overline{IKFE} + \overline{WEFHG} - \overline{WGHD'C} - W\overline{CDKI}.$

By Law of Conservation of Energy we have:

Net heat supplied = Net work done, or,

$$QA - QB = \overline{WIKFE} + \overline{WEFHG} - \overline{WGHD'C} - \overline{WCDKI}.$$

The original question to be answered was "How much work can be obtained from heat alone by means of an engine repeating a regular succession or cycle of operations continuously?"

In order to answer this question it must be demonstrated that the engine has received net heat and has delivered net work, to a reservoir outside of the engine and its operating substance.

Referring to the above equations, it is necessary that for the engine to have received heat that QA — QB must have a finite value, and that the work during expansion stroke must be greater than the work during the compression stroke.

Between condition at minimum volume QK and at maximum volume QH we have the following energy changes in operating substance:

*During Expansion.*

Received from A = QA.

Received from operating substance QK — QH.

*During Compression.*

Work, compression from *gh* to *cd* = QD′ — QH.

Work, compression from *cd* to *ik* = QK — QD.

(9) Therefore QA + QK — QH = QD′ — QH + QK — QD.

(10) QA = QD′ — QD.

But, from the third operation we have equation (4)

(4) QB = QD′ — QD.

(11) Subtracting QA — QB = O and QA = QB.

Therefore (1) no *net heat* has been received from a source outside of engine and operating substance, and (2) Quantity of heat received equals quantity of heat rejected.

During expansion total work done equals,

$$\overline{WIKFE} + \overline{WEFHG}.$$

During compression total work received equals,

$$\overline{WGHD'C} + \overline{WCDKI}.$$

Net work equals,

$$\overline{WIKFE} + \overline{WEFHG} - \overline{WGHD'C} - \overline{WCDKI} = KXD - XD'F.$$

But, Net work = Net heat.

Therefore $QA - QB = KXD - XD'F$, which in accordance with equation (11) equals zero.

Therefore

$$KXD - XD'F = O$$

$$KXD = XD'F \text{ and}$$

(1) No net work was delivered by engine.

(2) Work of compression equals work of expansion.

Therefore

1. A heat engine receiving heat only from an external source and delivering net work to an external receiver, and

2. Operating on a Carnot's Cycle consisting of (*a*) isothermal expansion at maximum temperature of cycle,

(*b*) Adiabatic expansion to minimum temperature of cycle,

(*c*) Isothermal compression at minimum temperature of cycle,

(*d*) Adiabatic compression to maximum temperature of cycle and to original condition of operating substance considered relatively to temperature, density and mode of aggregation, and

3. Performing these operations in a regular cycle or succession continually, is an impossible engine and cycle of operations for converting heat only into work.

However the engine has operated and has received heat and converted it into work during expansion of operating substance and during compression the engine has received work and converted it into heat.

But the heat received was at a high temperature and the heat rejected was at a low temperature.

Therefore the net result of a heat engine operating on a Carnot cycle is to receive heat at a high temperature and reject it at a low temperature.

And even this result is based on an assumption that Carnot neglects to mention, viz: that the operating substance has pressure in excess of the pressure on the opposite side of the piston.

If the opefating substance at "K" had no excess pressure (*i. e.*, greater than the pressure on the opposite side of the piston) and was at temperature of source of heat A, there would be no expansion or compression, no heat received or rejected, and area KFHD would become a point equal to zero.

# NOTES.

## THE FRENCH BRIEY-LONGWY IRON ORE BASIN.

We have received a copy of the translation of a secret memorandum which was addressed at the close of 1917 to the German Imperial Chancellor, Count von Hertling, and to Field-Marshal von Hindenburg, by Law Councillor W. Meyer and Dr. J. Reichert, on behalf of the Association of German Iron and Steel Manufacturers, and by General Manager Voegler and Dr. Ing. O. Petersen, on behalf of the German Ironmasters. Dr. Petersen is also the editor of the German periodical "Stahl und Eisen." The memorandum states: (1) That Germany's dependence upon foreign countries for its supply of iron ore constitutes a very grave danger to German industry and to the German State and people. It adds, (2) that forethought for the future renders the displacement of the Lorraine frontier necessary and unavoidable. In the third place, it enlarges upon the point that the mining lands (in France), which it is proposed by Germany to annex, are of enormous value for Germany's national welfare and for her waging of "a future war." A fourth and final chapter is entitled "Conclusion." The whole of the memorandum is very ably analyzed in a Foreword by the translator.

The memorandum states that the iron ore supplies to Germany's enemies are fully assured, whereas the German ironfields will soon be exhausted. Great Britain, it adds, imports a large quantity of ores, and before the war Germany also obtained foreign ores from the same countries which supply Great Britain; but, the memorandum further states, there is a great difference in this respect between the situation of Great Britain and that of Germany, and it arises from the British mastery of the seas. The huge navy of Great Britain ensured for that country, even in time of war, its overseas imports, whereas Germany was at once cut off from almost all the other ore-producing countries. The foreword deals with the comparisons made in the memorandum between the German iron ore resources and those of France, and calls attention to the fact that the German authorities do not mention the Bavarian iron ore deposits, estimations concerning which were given in the German journal "Glückauf" as late as March 12, 1910 ("Die Eisenerzvorkommen in der Fränkischen Alb"), when those deposits were calculated to contain 1,700,000,000 tons, a figure which certainly represents a minimum. The translator adds in this connection that the four German specialists, the authors of the memorandum, exaggerate both the annual consumption of iron ore in Germany and the importance of the French Normandy iron ore deposits, with a view to demonstrate that France, even were she compelled to give up to Germany the Briey-Longwy basin, would still remain in possession of very great iron ore resources. It should be remarked here that the memorandum states most candidly that "few people to this day are aware that Normandy possesses such rich ore deposits, for the German prospectors kept secret from everyone the knowledge they had acquired in the very last years before the war." The Germans were not always so very pessimistic in regard to their mineral resources, for in "Gemeinfassliche Darstellung des Eisenhüttenwesens" (5th edition, 1903, the last edition we have received), a book issued by the Association of German

Ironmasters, the latter, after reviewing the mineral resources of the whole world, congratulate themselves on the favorable position of Germany. The last three lines of the book state the following (translation): "* * * Under all circumstances, the prospects of Germany are more favorable than those of Great Britain, and this applies both to our (the German) coalfields and to the occurrences of ore." Since these lines were written, Germany, it is true, has lost Alsace-Lorraine, but she has ascertained the value of the Bavarian deposits above referred to.

The memorandum throws emphasis repeatedly upon the necessity of shifting the Lorraine frontier farther West owing to the German need of iron ore and to the proximity of many new and immensely important German establishments. It does this after comparing, as we have said, the mineral resources of Germany with those of "Germany's enemies," and then it states, practically in the light of an afterthought, that "by annexing all such necessary frontier areas, there would be incorporated in the German Fatherland a territory which for centuries belonged to the former German Empire." This means that if the Briey-Longwy district were absolutely barren and contained no ironstone in its subsoil, the German ironmasters would not think of annexing it, although it might have belonged formerly to the German Fatherland. Or they might have coveted it simply to increase the distance between France and the German establishments. Germany's "historic right" to the districts in question in this latter case still remains a secondary argument on the part of the German ironmasters who, it would seem, should have based their claim first and foremost upon the standpoint of history. Their brief historical data go as far back as Charlemagne; but the latter is quite as well known in French history as in German history. Charlemagne, in 800, was made Emperor of the West by Pope Leo III, and among the tribes he conquered were the Bavarians and Saxons. By invoking Charlemange, France might possibly lay claim to a part of the present Germany.

The figures quoted for the production of steel in Germany in 1913, the last complete peace year before the war, are as follow, the statements we give in brackets being taken from "Stahl und Eisen": Thomas steel (basic converter steel), 10,630,000 tons, Siemens-Martin steel, 7,977,000 tons. (This is probably made up of open-hearth steel, basic and acid 7,610,000 tons, and steel castings 362,916 tons.) Other steel is given as having reached 328,600 tons (probably made up of crucible steel 99,000 tons, electric steel 77,000 tons, and Bessemer steel 155,000 tons).

The figures for 1916 are also quoted, and are the following:

| | Tons. |
|---|---|
| Thomas steel | 7,654,000 |
| Siemens-Martin steel | 8,055,000 |
| Other steel | 473,300 |

For both years the figures cover Germany, Alsace-Lorraine and Luxemburg.

The fact that so much basic converter steel should have been manufactured in Germany in 1916, at a time when the country was unable to meet any of its own commercial needs and unable also to export, the whole activities of the nation being exerted in meeting the war requirements, is striking and should not be lost sight of. The output of open-hearth steel is practically the same in 1916 as in 1913. The memorandum states that the quantity produced in 1916 would not have reached the figure for 1913, owing to the lack in Germany of resources in discards and scrap, and owing also to the great shortage of labor. "Fortunately," it adds, "the successes of our army enabled us to secure from the conquered territories, besides great quantities of pig-iron, semi-finished and finished products, a large amount of discards and scrap. In this, we were helped, among

other things, *by pulling down the industrial establishments already devastated by war or threatened with destruction.*" The italics are ours. We may remind our readers that one of the addressees of the secret memorandum in question was Marshal von Hindenburg, who, we should think, was, better than its authors, conversant with the pulling down activities of the German Army. As it is, the total steel output of Germany was 18,935,600 tons in 1913 and 16,182,000 tons in 1916; the latter figure would hardly point to the alleged "great shortage of labor."

Reverting to the figures for output quoted in the memorandum, it may be surmised that in 1916 no armor plates and very few large-caliber guns, if any, were built for the German Navy; probably no guns, whether for the German Navy or Army were then built of crucible steel. We have always doubted the German assertion that their guns of recent construction were invariably built of that steel, considering the quality now obtainable in the open-hearth furnace. Krupp guns were manufactured before the war at the Cockerill Works, Seraing, and at the Skoda Works, Pilsen. The former works have been completely dismantled, no doubt because they were "threatened with destruction." An official of one of these two works asked whether he built Krupp guns of crucible steel, replied emphatically in the negative; the other firm used occasionally to purchase large naval gun tubes from outside steel makers, who may be said to manufacture tons of open-hearth steel for every ounce of crucible steel they produce. We believe that several Continental works still continue making crucible steel and also puddled iron. They, of course, find use for both metals occasionally, but their main object may be said to be the training of experienced workmen on whom they can rely in an emergency.

We wish to correct an error which has crept in the translation of the memorandum, on page 36. The authors are made to say: "* * * This was more notable from the end of 1870, when German engineers succeeded in putting to practical use the English discovery of the Thomas process * * *." The passage should read: "* * * This was more notable towards the end of the 'seventies" and so forth. The German writers could hardly claim to have improved a process before the discovery of that process. We do not know of any such German improvement of the basic process. In regard to the early statistics of basic and other steel output, they will frequently be found to cover, besides Germany, also Luxemburg and Austria. Even then, in the early 'eighties, the figures of output given in German journals do not show sufficiently high totals to point to any German improvement of the British process in question. The German ironmasters, like the ironmasters in every other country, had to overcome the difficulties inherent to the manufacture of a converter lining, adequate in every respect, before they could make a success of the basic process. The authors cannot have had in mind the "Scheibler process"; this, so far as we can trace, was patented in Germany in about 1886, and it was claimed to require a lesser lime addition, to reduce the duration of the blow and to give a better yield in steel and in phosphoric slag. It was adopted by some German firms and was dropped by several. But, given the British basic process, or, we should say, when a firm had acquired the right against the payment of a royalty to use that process, it would seem to us that that same firm was at liberty to vary at will the addition of baked limestone to suit its own particular requirements and conditions, and this variation would not appear to be patentable, the addition of limestone forming part of the original British invention. Be that as it may, the statement that "German engineers succeeded in putting to practical use the English discovery of the Thomas process" was one which was sure to appeal to von Hertling and von Hindenburg, who probably were not so well versed in steel manufacture as in other subjects. One of our foremost authorities in the manufacture of steel, Sir Robert

Hadfield, stated at the meeting of the Faraday Society on November 12 last, that "German metallurgists * * * have only copied, absorbed and made use of the principles of French, British, American and Swedish metallurgists. I cannot remember any specific instance of the Germans teaching us any basic principles in ferrous metallurgy."

The memorandum also lays great stress upon the very close connection which exists between the steel industry and agriculture, upon the value to agriculture of the phosphates obtained by the basic process. In our present brief review, this point may be viewed simply in the light of a corollary, and we need not refer to it specially.

It mentions "the future war" *ad nauseam* and states that "the displacement of our Lorraine frontiers is absolutely indispensable, not only for the security of the German Empire in a future war, but also for the consolidation of our national welfare * * *" Since Germany not only has not obtained the French Briey-Longwy basin which she has coveted for many years past, but has also lost the portion of that basin which is situated in the provinces she wrested from France in 1871, it would therefore follow that "the future war" now becomes to her an impossibility. But there remain to her the above-mentioned Bavarian deposits of 1,700,000,000 tons, upon which she appears not to have laid much stress hitherto, and the Entente Nations would still have to view contingencies with great caution. The question of "the future war" is outside our province, it is one entirely for the Peace Conference to deal with, but we may be allowed to connect the matter with a statement made many years ago in quite different circumstances by a learned British judge. He said: "The harness of certain horses contains a special strap, called the 'kicking strap.' The horse does not know the strap is there until it wants to kick; and when it wants to kick, it can't." We are quite content to leave all these questions in the hands of the Entente diplomatists who, we feel sure, will do the needful.

In conclusion we recommend the report to our readers. The English version is published in pamphlet form, under the title "German Designs on French Lorraine," at the price of 6*d*. net, by Messrs. George Allen and Unwin, Limited, Ruskin House, 40, Museum-street, W.C. 1.—"Engineering."

## IMPROVEMENTS IN THE CONSTRUCTION OF SHIPS.*

The idea of providing a central pipe or duct in the lower portion of a vessel for the purpose of dealing with fluid distributed in different compartments along its length is not new. It is proposed in this particular development to provide the central fore-and-aft duct by dispensing with the ordinary vertical keel plate, and by working instead a relatively narrow box keel running the length of the vessel, and having an uninterrupted passage through it from one end to the other.

Dealing first with the form of this duct keel and its relation to the adjacent ship structure, reference may be made to Figs. 1 and 2.

Fig. 1 represents, in cross section, an ordinary type of vertical keel structure. Fig. 2 represents sections across the proposed duct keel, the framing shown on either side of the keel in Fig. 2 being of the plate-frame type. The maximum width between the vertical plates is not at present intended to exceed 2 feet 6 inches, while it may be as little as 1 foot 9 inches in particular cases. The depth of the duct would generally be the normal full depth of the vertical keel.

*Abstract of paper read before the Institution of Engineers and Shipbuilders in Scotland, by Mr. E. F. Spanner, R. C. N. C.

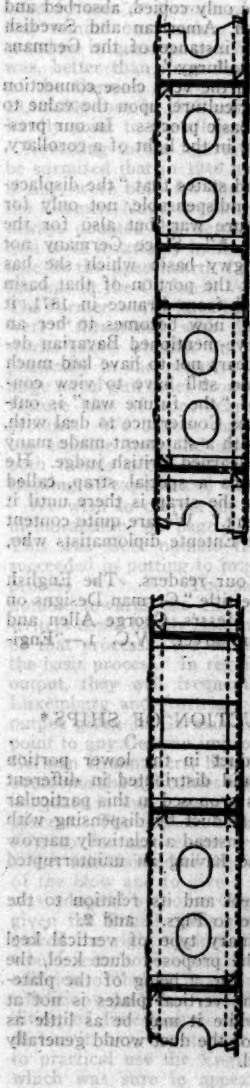


Fig. 1.—Ordinary Form of Vertical Keel.

Fig. 2.—Proposed Form of Duct Keel.

A=Valve in side of Duct. B=Pipe through Duct. C=Valve for Emergency Trimming. D=Divisional Valves.
E=Suction for Salt Water or Oil Fuel.

Fig. 3.—Proposed System of Valves used in conjunction with Duct Keel.

It is at present suggested that in general the poundage of each of the vertical plates forming the duct keel in any vessel should be not less than two-thirds the poundage of the single keel plate proper for that particular vessel, that the poundage of the gutter plate should be 20 per cent in excess of the usual poundage, and that the angles should be not less than 6 inches by 6 inches by ½ inch in section. It is not proposed to increase the thickness of the keel plates, nor is it proposed to fit any stiffening to these keel plates inside the duct. If necessary, stiffening can be fitted across the upper portion of the duct, to support the gutter plate in wake of machinery weights, etc., or for such other purposes as may be desirable.

The actual building of such a duct keel structure as that outlined above calls for few remarks; it is a relatively simple proposition, particularly as this structure would be the first portion of a vessel to be tackled. Sufficient room for riveting is provided, even if the width of the keel is only 1 foot 9 inches, while with a width of 2 feet 6 inches the men would be far from cramped. It would be preferred that the top and bottom angles and the laps and butts of plating should be riveted, and that, after riveting, the edges should be electrically welded instead of being caulked. This would ensure the duct being thoroughly tight and capable of withstanding a good head of fluid. If welding plant is not available, however, a satisfactory job can be secured by proper caulking.

The longitudinal strength of a vessel fitted with a keel structure as proposed is obviously greater than if the ordinary vertical keel is fitted, particularly against hogging stresses.

So far as docking stresses are concerned, if the vessel is properly docked, the members of the keel structure are less severely stressed than is the case with a single vertical keel plate. If she is docked off the center line an unfair stress is brought upon the keel structure, but this is less severe in the case of the box keel than with the ordinary keel. As regards transverse strength, objections may be raised to accepting a box-keel structure without internal transverse-framing, and, as it is desired to dispense with such transverse framing as far as possible, it will not be out of place to deal with the question of transverse strength at some length. In the first place, there is no appreciable reduction in the form-retaining value of the transverse bulkheads, due to the loss of the small area of plating represented by the cross section of the duct keel.

Taking a short transverse section of the vessel, it is generally agreed that the function of the transverse framing is to act as the web of a girder of which the two flanges are the inner and outer bottoms respectively. This girder affords the stiffness necessary, in addition to that provided by the bulkheads, to enable the vessel to maintain her shape. The forces on the bottom of the vessel generally tend to push the bottom into a shape convex from above, *i.e.*, they put the top member of the girder in tension and the bottom member in compression.

It is proposed in the new form of construction to connect the port and starboard portions of the transverse girders to the rigid box forming the duct keel. Constructed in the manner shown in Fig. 2, the duct keel would not collapse except under the effect of stresses which would more than equally injure the structure of a vessel of ordinary construction, and certainly not under any stresses which would normally be transmitted to it through the adjacent structure. If for any particular reason it is desired to fit transverse stiffening to the upper portion of the duct, this is rendered an easy matter by working the upper angles outside the duct.

The sketches and explanation above refer only to the case of a vessel having a double bottom built on the transverse system. The idea is equally easily applied to a vessel designed on the Isherwood system.

It is now proposed to outline the services to which the duct keel can be adapted, considering first the case of a vessel burning coal and using the double-bottom tanks primarily for the carrying of water ballast. In the ordinary way a vessel of fair size would have from 12 to 20 or more double-bottom tanks, and for working them would require a suction leading into each tank. These suctions would each start from a valve box or boxes in the machinery space, and would either run forward and aft above the inner bottom, piercing one watertight bulkhead after another until over their respective tanks, where they would be turned down and led into them through the inner bottom, or else would run through the double-bottom spaces. Such systems necessitate many hundreds of feet of suction pipes and large numbers of holes in the watertight bulkheads or frames and in the inner bottom. Cargo stowage is lost owing to the necessity of protecting these leads of pipes, and constant care and attention is necessary to provide against deterioration. If the vessel is bilged, and damage results to one or more of the suction pipes, special precautions are necessary to prevent water from finding its way through these pipes into compartments far away from the region of the actual damage, and so further jeopardizing the safety of the vessel.

Fig. 3 illustrates the proposed system of dealing with water ballast by means of a duct keel. The duct runs fore and aft between the port and starboard double-bottom tanks, and each tank can be put into connection with it by opening a valve in the side of the duct. These valves are marked A. The duct keel is actually a long double-bottom compartment, and is provided with the usual air escapes. It is proposed that the tail pipe of the ballast pump should have two branches—one leading to the forward end and the other to the after end of this keel compartment. These are shown in the sketch running forward and aft from the suction at E. By using the one or the other, according as to whether the vessel is trimming by the head or by the stern, it would be possible to completely empty any tank or tanks. To fill any tank it is only necessary to open the valve putting this tank into communication with the duct keel and to pump water into the latter until soundings show the tank to be full.

It will be seen that the provision of the duct enables virtually the whole of the ballast-suction system to be dispensed with, while no holes are necessary through watertight bulkheads. Further, water entering the ship following damage cannot find its way along the ship to other undamaged compartments by way of the duct keel, except under the control of the ship's officers. By the adoption of a simple type of pressure-operated valve, control of all the valves on the duct keel can be, if desired, centralized in the engine room, the necessary small pressure leads running along inside the duct.

Returning to the question of damage resulting to a vessel from collision or similar mishap, it will be generally admitted that, of all portions of a ship's structure likely to be damaged at sea, the least likely to suffer is certainly the keel structure. Except grounding, there are probably few cases in which damage has resulted to the vertical keel, and, as the proposed box keel is certainly much stiffer than an ordinary single-plate keel, it is not unreasonable to expect that, even should the vessel take the ground, the duct keel would remain substantially intact. However, to provide against a single damage putting the duct out of operation, two simple-type vertical doors or valves are provided, dividing the duct into three sections. It is not essential that these doors should be more than roughly watertight. They are shown marked D on Fig. 3. If it is

desired to use some of the double-bottom or peak tanks for fresh water or for feed water, the suction pipes to these tanks may be led along from the pump to the tanks by way of the duct keel, thus avoiding the piercing of any watertight bulkheads for the passage of these pipes. This is illustrated in Fig. 3, the pipes being marked B.

Consider now the service which can be obtained from a duct keel in the case of a vessel fitted for burning oil as fuel, and carrying this oil in the double-bottom compartments and in some of the holds. As in the case of the vessel first described, the duct keel could be put into communication with any of the double-bottom compartments containing the oil fuel by the opening of the proper valves, while, by fitting valves in the roof of the duct, it would be possible to put the duct into communication with holds in which fuel oil might be stowed. There are certain points in which the problem of dealing with oil fuel differs from that of handling water ballast, such points arising principally owing to the viscosity of oil at low temperatures. This viscosity is generally so great that it is impossible to pump the oil from the tanks until by some means the temperature of the oil has been raised sufficiently to permit of a considerable lowering of the viscosity. The means usually adopted to heat the oil in the tanks is to fit grids of steam pipes, such grids being necessary, one in each of the tanks used for carrying oil. This is both an expensive and uneconomical procedure, but so far no other means have been found of overcoming the reluctance of the oil to being drawn up through the suction pipes. That the usual steam-heating method is uneconomical is obvious when one considers the fact that, while it is necessary for the oil-fuel suction to be practically at the lowest point of the compartment, the oil itself as it is heated rises to the top, and not until the whole volume in that compartment has been raised in temperature is it generally possible to start a flow which is at all likely to be consistent. Further, only one section of pipe is used at a time, so that 90 per cent or more of the heating system is always out of action and represents useless loss of deadweight. It is further necessary to carefully lag all the suction pipes to prevent stoppage of flow due to cooling of the oil while actually in these pipes, and it is common practice to lead a steam pipe along with the suction pipe, lagging being bound around the two pipes.

It is claimed for the proposed duct-keel system that not only is it possible to eliminate practically the whole of the pipes forming the ordinary filling and suction systems, but also that practically the whole of the heating pipes can be dispensed with. Dealing with the latter claim first, attention is drawn to the following considerations:

1. That the duct keel lies at absolutely the lowest point of the ship, and that, therefore, if easy access is provided from the double-bottom compartments or holds to this portion of the ship, gravity will ensure that flow will result into the duct keel, even if the oil be only semi-fluid.

2. That if steps are taken to heat the oil in the duct keel by means of the usual steam pipes, this oil must necessarily attain a temperature sufficient to enable pumping operations to start much earlier than would be the case were the oil in an ordinary double-bottom compartment, owing to the fact that once the oil has entered the duct it is virtually trapped there, while the volume of oil contained in the duct is comparatively small. Further, as the duct will always contain heated oil when in use, flow from the double-bottom compartments or holds to the duct will be facilitated.

3. It will be quite unnecessary to lag the suction pipes, as these will be extremely short outside the duct, the two tall pipes of the pump leading direct into the duct through the inner bottom.

It will be seen that by fitting heating pipes only to the duct keel, it will be possible to handle the oil effectively, and the very large expense involved in providing for heating pipes to all tanks will be obviated.

As regards the suction arrangements, these will consist of tail pipes from the one or more fuel pumps fitted, and the long leads usually necessary to the several tanks can be dispensed with. Similarly, it will be possible to dispense with the filling pipes to each of the oil-fuel tanks, these tanks being filled by pumping oil into the duct keel and opening the valves to the various tanks in succession.

As has previously been pointed out when discussing the use of the duct keel for water ballast, damage to the duct is unlikely, but if it should occur either forward or aft, the damaged section could be isolated by shutting one or other of the transverse doors. Should some very extensive damage quite put the duct keel out of commission, fuel could still be obtained for a restricted radius of movement from the settling tanks, which would normally contain a certain quantity of oil.

A point worthy of notice is that no loss of double-bottom stowage results from the introduction of the duct keel, as, whether the vessel carries water ballast or oil in the double bottoms, she can, when loaded, carry the duct full of either water or oil, as the case may be. As the duct differs from an ordinary double-bottom compartment only in that it is deeper and freer of access, no difficulty should arise in clearing it of fluid for purposes of inspection, etc.; in fact, it should be a simpler matter to clean out the duct than to clean out an ordinary compartment.

It is now proposed to indicate a further direction in which the new form of keel is likely to be of very considerable value.

The duct keel is of value in the saving of vessels which have suffered serious underwater damage by what is known as the Brunton system. The Brunton scheme contains two main ideas:

1. The addition of water to the damaged vessel so that the maximum corrective effect is obtained from every ton of water added.

2. The utilization of existing powerful pumps or of pumps specially introduced to pump the necessary weight of water rapidly into position.

To carry out these ideas it is first necessary to provide that the bulkheads limiting what may be called the emergency-trimming compartments should be watertight and specially stiffened. This is not difficult. It is also necessary to arrange for some connection between the powerful pumps, which would be situated in the machinery compartments, and those emergency-trimming compartments. This Messrs. Brunton have hitherto proposed to do by fitting large pipes running from the pumps to these compartments. The objections to such a proposal are serious—from the naval architect's, builder's and owner's points of view. Such pipes, to obtain the required rate of discharge, must be 12 inches or more in diameter, and they form a source of weakness in the completed vessel, are difficult to arrange for, and are somewhat costly to instal and maintain. It is these several objections that have undoubtedly somewhat hampered the development of this very valuable system.

The manner in which all these objections are met by the utilizing of the duct keel for the purpose of communication between pumps and emergency-trimming compartments hardly needs explaining. The duct keel is in reality a rectangular pipe of very large cross-section, running from one end of the vessel to the other, and to provide communication between it and the forward and after emergency-trimming compartments it is only necessary to fit a suitably operated valve or door at each end. It is an equally simple matter to provide the necessary connection from the powerful pumps to the duct, whether it is decided to utilize the circulating pumps, as has been proposed, or whether other and more flexible pumps are fitted for the purpose.

By the provision of short transverse connections in the double bottom between the duct keel and suitable wing compartments, arrangements could be made to pump water into what might be termed "emergency-keel compartments" for correcting heavy list due to damage, the correction of list being also a feature of the Brunton system.

Reference may be made to the fact that the duct is practically available for use in an emergency, even if it be itself badly damaged. Suppose, for instance, that the fore end is crippled when the forward compartments are bilged. In the first place provision is made for cutting off this portion of the duct from the remainder by a door or valve in the duct, but even if that door or valve is damaged and unable to be closed, it is an undoubted fact that the head over the end of the duct at the undamaged end of the ship is very much less than the head over the damaged end of the duct, so that, by starting the pumps, water is bound to be forced into the emergency-trimming compartment aft, with a beneficial effect on the trim of the vessel.

Another point which may be emphasized is the fact that the duct can be used for the Brunton system, whether its normal service is that of filling and emptying water-ballast tanks or whether it is used for working oil in a vessel fitted for burning oil fuel. Suppose the duct to be full of oil when the vessel is bilged. It is of little consequence whether the fluid pumped into the emergency-trimming compartment is sea water or sea water and oil mixed when the actual safety of the ship is concerned, and there is no reason for varying the procedure to be followed on account of the presence of oil instead of water in the duct.

An effort has been made in the foregoing to touch upon most of the principal features of the form of keel construction recently patented by Mr. J. H. Silley, of the firm of R. & H. Green, and Silley, Weir, Ltd., jointly with the author, but there are a number of points in favor of the idea which could profitably be dealt with at greater length, and others which have not been mentioned.—"Shipbuilding and Shipping Record."

---

## THE CONSTRUCTION OF SHIPS' LINES.

In a valuable paper on "The Design of Lines and Some Considerations of Form," read before the Newcastle Branch of the Association of Engineering and Shipbuilding Draughtsmen, Mr. R. Allan, B.Sc., includes some useful information in regard to methods facilitating the preparation of ships' lines from previous successful designs.

Fig. 1 shows a graphical method of constructing a body plan to give a required displacement when the new vessel is to have the same block and mid-area coefficients as a basis vessel. The method of procedure is first to draw the center, half-breadth, base and load-water lines for the new design on transparent paper or cloth. Then draw in water-lines corresponding to the water-lines of the basis ship at a distance apart given by original spacing multiplied by draught of new ship divided by draught of basis ship. To obtain the offsets for any given water-line, say $W_1L_1$, place the new body-plan outline over the basis ship body-plan so that points O on corresponding water-lines coincide; then slew upper plan round until the half-breadth spot, $W_1$, on the new design lies on the half-breadth line, $W_1W$, of the basis ship, as shown. Project down the various water-line spots of the basis ship as at R, S, B, etc., on the water-line $W_1L_1$ by lines $RR_1$, $SS_1$, $BB_1$, etc., parallel to $W_1L_1$ on the new design. The case illustrated is for a new design having a greater breadth

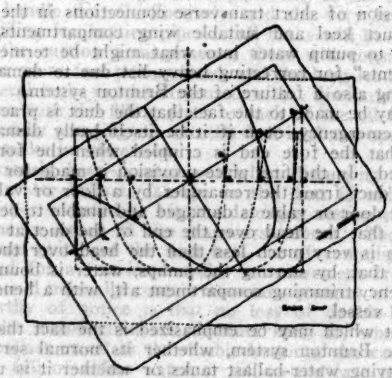


Fig. 1.

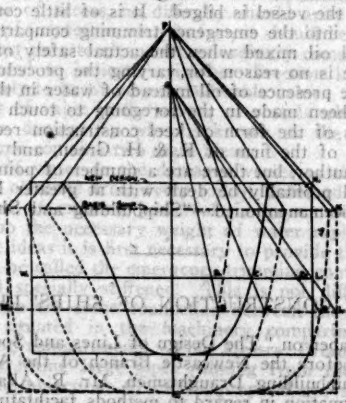


Fig. 2.

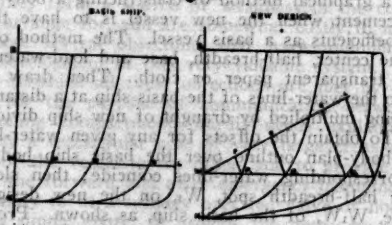


Fig. 3.

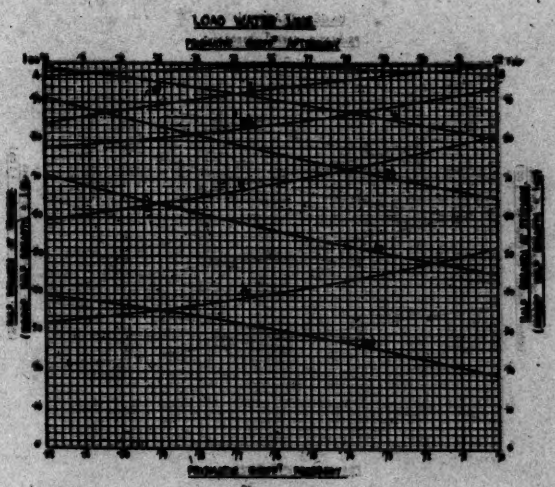


Fig. 4.

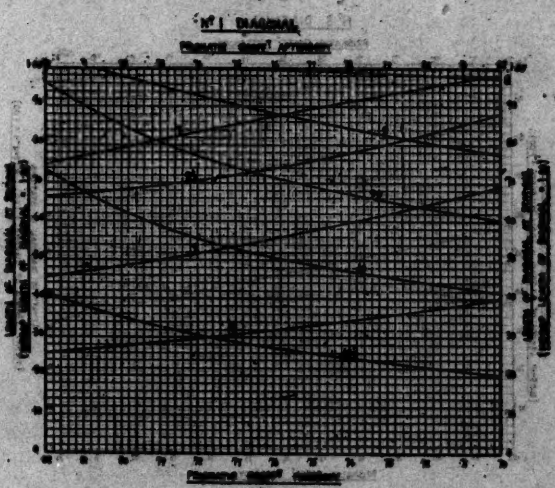


Fig. 5.

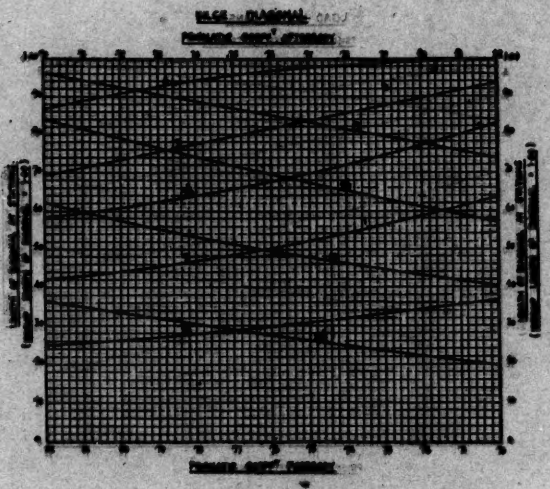


Fig. 6.

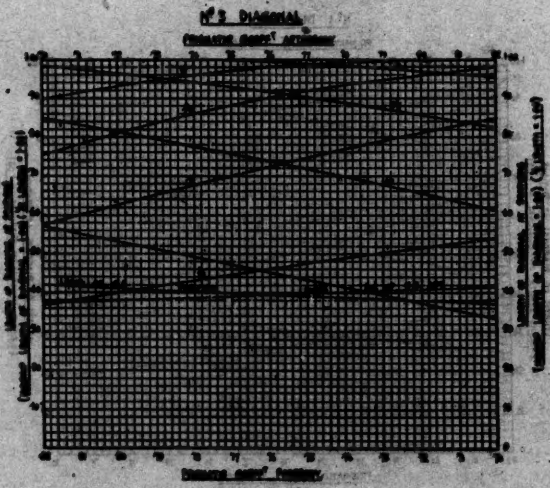


Fig. 7.

than the basis vessel, but if the reverse be the case the procedure is very similar, the body-plan of the basis ship being in that event placed in the inclined position.

Another graphical method of solving the same problem is shown in Fig 2. In this method the new outline body-plan is placed over the basis ship body-plan with the center-lines coinciding, so that corresponding water-lines WL and $W_1L_1$ are clear of each other by any convenient distance. To obtain spots on $W_1L_1$ for new design, join the spots $LL_1$ by a line which meets the center-line at O. Join OB, OC, OD, etc., to cut $W_1L_1$ in B, C, D, etc., respectively, which will be the spots required. For convenience in drawing, the angle WLO should be about 45 degrees.

Fig. 3 illustrates a graphical method of constructing a body-plan to give a required displacement, keeping the same form of sectional area curve as the basis vessel. This method is particularly suitable to the design of destroyers, cruisers, channel steamers, etc., where the sectional area curve of the basis vessel has been found a very good one. The displacement and draught being fixed and therefore the block coefficient, the same prismatic coefficient and therefore the same sectional area curve can be kept by simply modifying the mid-area coefficient. As an example, suppose the basis ship to have a block coefficient of .60, mid-area coefficient of .88 and prismatic coefficient of .682, and the block coefficient required

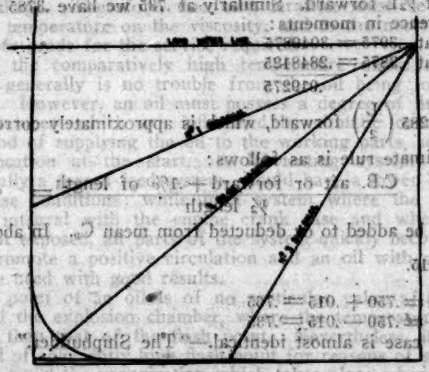


Fig. 3.

for the new ship to be .59. The mid-area coefficient for the new ship to keep the prismatic coefficient the same as for the basis ship must therefore be $\frac{.59}{.682} = .865$. To draw the new ship's body-plan, first design the mid-ship section to the desired shape and coefficient. Then space off water-lines corresponding to the water-lines on the basis ship's body-plan in the proportion of the draughts of the two vessels, *i.e.*, in the proportion $\frac{K_1D_1}{KD}$. Next, to find spots for any water-line $W_1L_1$ corresponding to WL in the basis ship, alter the offsets of the basis ship in the proportion $\frac{W_1L_1}{WL}$. This may be quickly performed graphically as shown in Fig. 3. It should be noted that the offsets for any given water-line are altered in the proportion of the corresponding half-breadths of that water-line at the mid-

ship section, and not in the proportion of the half-beams of the ships as was done in the two methods previously described.

When limits of parallel middle body permit, the methods illustrated by Figs. 1 and 2 can by a re-spacing of the sections be extended to cover cases where the block coefficients are different in the vessels under comparison. The procedure in such cases has already been described in "The Shipbuilder."*

Offsets for standardized lines based on the experiments of Baker are given in Figs. 4, 5, 6 and 7. The offsets are for the load water-line and three diagonals placed as shown in Fig. 8. The prismatic coefficients for the fore and aft bodies respectively are selected to give the required longitudinal position for the center of buoyancy. To enable this readily to be done the positions of the centers of buoyancy for each body are included in Fig. 7. The method of procedure is as follows: Suppose, for example, ship 400 feet long, with center of buoyancy 2.6 feet forward, and total prismatic coefficient .750:

$\frac{2.6}{400} = .0065L$ forward, or $.0130\left(\frac{L}{2}\right)$

Try .765 forward and .735 aft.

Reading from curve of locus of C.B. forward at .765, we find C.B. to be .3975 of ½L forward. Similarly at .735 we have .3785 of ½L aft.

Find difference in moments:

| | | | |
|---|---|---|---|
| .765 | at | .3975 = | .3040875 |
| .735 | at | .3875 = | .2848125 |
| 1.500 | | | .019275 |

$\frac{.019275}{1.5} = .01285\left(\frac{L}{2}\right)$ forward, which is approximately correct.

An approximate rule is as follows:

$$\frac{\text{C.B. aft or forward} + .1\% \text{ of length}}{\text{½ length}} =$$

Amount to be added to or deducted from mean $C_p$. In above example:

$\frac{2.6 + .40}{200} = .015.$

$C_p$ forward $= .750 + .015 = .765$

$C_p$ aft $= .750 - .015 = .735$

which in this case is almost identical.—"The Shipbuilder."

## INTERNAL-COMBUSTION ENGINE LUBRICATION AND LUBRICANTS.†

By P. H. Conradson.

The main consideration in the selection of oil for the various classes of service are not well understood. Too much weight is given to vague inferences as to the relative value of viscosity and flash point alone, without taking into consideration many other important factors; and while it is true that some few engine manufacturers have conducted exhaustive investigations to determine just what particular grade of oil

*See article on "Some Technical Aspects of Shipbuilding Contracts," July number, 1915; No. 59, Vol. XIII.

†From a paper before the American Society for Testing Materials.

should be used by their engines, the greater part of the work of developing suitable lubricants for the various combustion engines has been done by the refiners and dealers of oil.

While the determination of viscosity—that is, body or consistency at a given temperature—is one of the vital tests usually applied to lubricating oil for internal-combustion engines, it is of importance only when the user of the oil is thoroughly familiar with the characteristics of the different crude oils of which it is made and is sure that the mechanical condition calls for the use of an oil having the stated viscosity. Two oils of the same viscosity, made from different crudes, may behave quite differently under identical working conditions.

VISCOSITY TESTS AND TEMPERATURE.

The determination of viscosity of an oil, to be of the greatest value, should be made not only at the lower temperatures, but also as near the temperature of its use as consistent. On the one hand, an oil of low viscosity might be preferable, since it absorbs less power than a thicker oil in the separation of metallic surfaces moving at a high velocity over each other; in other words, high viscosity means a high internal friction. On the other hand, if the oil is too thin it is more easily displaced from between the bearing surfaces, and a heavier oil might then be preferable, as the engine will be more flexible at low speed, owing to better seal and less leakage. Therefore, due consideration must be given to the influence of temperature on the viscosity, and to the necessity of selecting the proper grade for the service required.

Owing to the comparatively high temperature under service conditions, there generally is no trouble from the oil being too thick when once in use. However, an oil must possess a degree of fluidity at ordinary temperatures—which are influenced by climatic conditions—suited to the method of supplying the oil to the working parts, so as to obtain proper lubrication at the start. A lubricating system with exposed pipes, especially a forced-feed system, should have a lower "cold-test" oil to meet these conditions; while in a system where the oil is carried in a pump integral with the engine crank case and where the supply pipes are not exposed, all parts of the system quickly become sufficiently heated to promote a positive circulation and an oil with a higher "cold test" may be used with good results.

The flash point of an oil is of no particular value after the oil has once entered the explosion chamber, where the temperature is considerably higher than that of the flash point. Nevertheless, it is important to use an oil of sufficiently high flash point for reasons of economy if for no other, to resist the vaporization which takes place when the oil comes in contact with the highly heated surface of the piston head, and other parts below the piston. In considering this it is well to remember that with many oils there is little connection between the evaporation (heat) loss and the flash point. Investigation has shown that while two oils may have the same flash points, the percentages of evaporation loss at a given temperature may be quite different. Generally speaking, however, the evaporation loss is greater in oils having a low flash point.

Obviously, that oil which, when subjected to an evaporation test at a certain temperature, loses the least and leaves a residue that has been broken up or altered in character as little as possible by the heat treatment is the most desirable for lubrication of internal-combustion engines. Therefore, the evaporation and oxidation tests offer a promising means of examining the oil and noting the changes that develop in different oils under similar conditions of treatment.

Oil consists principally or entirely of compounds of carbon and hydrogen. At high temperatures these hydrocarbons decompose, forming volatile combustible gases and heavy hydrocarbons, or carbonaceous matter, and free carbon. This breaking up may be due to improper methods of

refining or to the nature of the crude oil. Only the most highly refined and filtered oils should be recommended to meet these conditions.

Much has been written about the carbon residue of the oils for this class of service. The carbon in all oils can be fixed by driving off the oil vapors (without addition of air), leaving a layer of carbon deposit, called carbon residue. The condition of the carbon formation in the cylinder is somewhat different from that found in the carbon-residue test, due to the fact that in the cylinder the oil is spread out in a thin film on hot metal surfaces, and exposed to the burning gases of the explosion, which may carry widely varying amounts of oxygen to combine with the oil. Also, the amount of carbon deposited in the cylinders is governed by the amount of oil reaching the explosion chamber, and may depend to a great extent upon the mechanical fit of the pistons, piston rings and cylinders. It is expected that some oil will find its way into the combustion space, but it is assumed that it will be burned up without depositing an excessive amount of carbon on the cylinder walls and pistons.

One of the most injurious effects of improper lubrication is the formation of this carbon deposit around and under the piston rings. Such a deposit soon renders the spring rings inoperative; they become partly fastened in their grooves, and in this condition form one of the most prolific causes of cut and scored cylinders and broken piston rings. One of the causes of this trouble may be the use of too much oil. From the result of much investigation along this line it has been conclusively demonstrated that an oil with the lowest carbon residue, other things being equal, will leave behind the lowest carbon deposit in the explosion chamber.

In connection with the physical characteristics of the carbon deposit it is of importance to select a lubricant which leaves a loose and flaky or soft carbon deposit, easily removed, rather than one leaving a dense, hard deposit difficult to remove.—"International Marine Engineering."

## TOSI DIESEL ENGINES.

We have recently published on several occasions information indicating the rapid extension which is being made in Allied and neutral countries in the internal-combustion engine industry, especially where marine work is concerned, and in this respect the part being played by our Italian Allies is in keeping with their past record of substantial performance in the development of this type of prime mover.

Messrs. Franco Tosi have already built over 400 Diesel engines, these being of both the two-stroke cycle and four-stroke cycle types, totalling, in 1,200 cylinders, about 120,000 B.H.P. The range of horsepower developed hitherto per cylinder is from 15 B.H.P. to 400 B.H.P., and the average power per cylinder being 100 B.H.P., a high average, be it emphasized, considering that the engines comprise besides those used for land power generating stations, pumping sets, marine engines for cargo vessels, submarines, and many for auxiliary service on board ship. The two-cycle engines were built chiefly for the higher powers, i. e., over 200 B.H.P. per cylinder. The speeds of revolution range from 110 r.p.m. to 120 r.p.m. for land engines, to 400 r.p.m. for submarine and high-speed engines. Of marine engines alone Messrs. Franco Tosi have built, or have under construction, 50 engines, totalling 25,000 B.H.P., the largest size hitherto being a six-cylinder unit of 2,400 B.H.P. and the smallest 150 B.H.P.

Messrs. Franco Tosi have decided to concentrate on the four-stroke cycle engine in the future, for they are of opinion that if the same life

is to be obtained from the two-stroke cycle type as from its rival, any advantage of reduced space and weight which it may possess disappears. Advocates of the two-stroke cycle engine give, as one of the principal reasons in favor of this type of engine, the greater power that can be developed in a given size of cylinder, but Messrs. Franco Tosi point out that this advantage is only obtainable at the expense of the greatly increased temperature of the cylinder, piston heads, etc., which arises from the combustion of the larger quantity of fuel necessary for the increased power per cylinder. The ratio of increase in consumption of fuel per cylinder of equal size is, in fact, greater than the ratio of increase of the power obtained from the cylinder, the consumption of fuel per horse-power hour being somewhat greater in the two-cycle than in the four-cycle engine. This greater quantity of heat developed in each cylinder in a given time and the resulting higher temperature is confirmed by the color of the surfaces of the parts exposed to it, and is responsible for the troubles that have been experienced in the two-stroke engine.

It follows, therefore, that the two-stroke cycle engine, if designed with the same size of cylinder, will have a shorter life and may require more frequent overhauling than an engine of the four-cycle type. This is a fact of great importance not only with heavy oil engines for cargo-boat propulsion which must run uninterruptedly at long periods at full load, but also for the lighter type of engine used for submarines, which, although not required to run fully loaded for such long periods, are subjected to high stresses. Considerable progress has doubtless been made with the two-cycle engine, and many improvements introduced into the design, material, cooling of the cylinder and piston, to overcome the difficulties experienced. The modern two-cycle engine is thus undoubtedly more reliable than the older types, but such engines have been in service for too short a time for a definite judgment on them to be arrived at. On the other hand, rapid progress has also been made in the design of the four-stroke cycle engine, whilst many years of highly satisfactory service at full load have already been recorded. Regarded in another way, if an equal life is required from both types of engine—which is what is called for in both land and marine service—an equal quantity of fuel should in both types be consumed in a cylinder of a given size in order to secure equal temperatures. If this condition is adhered to the result will be that the same power will be developed by both types with an approximately equal weight and space.

The crosshead type of construction has been adopted by Messrs. Tosi for all slow-running engines, both for marine and land work, in lieu of the trunk piston design. The chief advantages of the trunk piston engine are simplicity, lightness and compactness. The piston, however, with this type is required to perform the double function—firstly, of maintaining airtightness between the high pressure of compression and combustion on the one side and the crank chamber at approximately atmospheric pressure on the other (*i. e.*, the piston is an obturator); and secondly, of a crosshead in taking the side thrust consequent upon the angularity of the connecting rod. The reliability of its performance in this dual rôle, unless with special attention, is not all that can be desired. When the rings lose their elasticity due to continued exposure to high temperature, or become gummed up, gases leak past the piston ring, and heat up the working surfaces of the piston with the possible result of piston seizure.

The proximity of the gudgeon-pin bearing to the hot piston crown tends, also, to overheat this bearing, giving rise in turn to the danger of distortion and seizing of the main piston. Further, leakage of burnt oil past the piston rings contaminates the crank chamber lubricating oil, causing a more speedy reduction in its lubricating qualities than would arise by normal use. The consumption of lubricating oil, also, is high, due to

the oil which is splashed on to the cylinder walls finding its way into the combustion chamber and so out through the exhaust.

In the present article we propose to deal with the standard Tosi slow-speed four-stroke cycle engine, now being constructed for land and marine work, of various sizes of cylinder units, three, four, six and eight units going to make up the complete engine, according to the power required. For a detailed description reference will be made to Figs. 1 to 4. Of these views, Fig. 1 is a longitudinal combined elevation and section, while Figs. 2 and 3 are two cross-sections, one through a main cylinder and one through the compressor. Fig. 4 is a section, to a larger scale, through a main cylinder head, showing the inlet and exhaust valves.

As already stated, the engine is of the four-cycle type and the six cylinders are approximately 17 inches diameter by 26 inches stroke, developing 83.5 brake horsepower per cylinder, or 500 brake horsepower total in the six cylinders at 180 r.p.m. The piston speed at full speed of revolution is 780 feet per minute, and at full power the corresponding mean effective pressure is 62.5 pounds per square inch. With a mechanical efficiency of 78 to 80 per cent. this gives an indicated mean pressure of about 80 pounds per square inch, which is a moderate figure, allowing a very desirable margin for contingencies and making possible the development of the 500 B.H.P. well within the capacity of the engine at a lesser speed of revolution than the designed figure of 180. We hope to publish at an early date the result of trials at sea of engines of this type. The piston speed is also moderate. The stroke/bore ratio of 1.55 gives ample room for the accommodation of the valves in the cylinder head, keeps the height of the engine within reasonable dimensions (dismantling as will later be explained is largely accomplished from above through the cylinder), and makes possible, if required, the built-up type of construction for the main crankshaft.

The engine is of the slow-speed, enclosed, crosshead, forced-lubrication type, and presents a number of novelties in design. The main framing follows land practice for Diesel engines, and marine practice for steam engines, in that each cylinder with its framing is a separate, interchangeable unit. The jacket, framing, guides and the columns are cast in one piece of grey cast iron of sufficient scantlings to carry the main tension stresses of the piston load. Each unit is bolted up fore and aft, and the whole makes a most rigid structure, although somewhat heavy, since no through-bolts are utilized to take the tension load. The whole framing is enclosed by light metal doors.

Each cylinder and the cylindrical guide faces are machined at the one setting, as is quite common practice with land work, this making for speed of construction and ensuring concentricity. Into each outer cylinder casting a liner of special cast iron is pressed. These liners are of special design, having a spiral rib cast on the outside, in which the bosses for the cylinder lubrication connections are incorporated. The purpose of this spiral rib is to give to the cooling water passing through the jacket a definite path and high velocity, obviating pockets and ensuring equal cooling. The top of this rib is machined and fits into the jacket, so supporting the liner and reducing the thickness required. Each liner is anchored at the top end, and is free to expand at the lower, the water joint at the lower end being made in the usual way.

In regard to the design of the cast-iron framing and bed plate (3), these can be said generally to follow accepted practice. The bed plate (3) is framed of two side I beams, with box-section cross girders carrying the main bearings totally enclosed to form a sump for the lubricating oil, as is usual with forced-lubricating engines. The bed plate is not supported at the center, the cross girders being of ample section to transmit the loads.

To turn to the main running parts of the engine, the piston (4), piston rod, crosshead, connecting rod (5), and crankshaft (2). The piston is quite shallow, since there is no side thrust to be carried and it is fitted with ten Ramsbottom cast iron rings to ensure gastightness. It is connected to a flange on the top end of the hollow piston rod, and is designed for oil cooling. Earlier Tosi engines had water cooling of the main pistons (see "Engineering," July 24, 1912), but oil cooling with this special design of piston is now standardized. In the piston crown is a spiral passage for the cooling oil, which is delivered to the center of the piston crown—the hottest part—through a central pipe in the hollow piston rod, then spirals round the crown and discharges out of the piston crown down the outside of this hollow pipe and so to the crankchamber. It is claimed for this method that the efficiency of cooling is exceptionally high, and so, in spite of the low specific heat of lubricating oil as compared with water, satisfactory results are achieved, and the risk of carbonization of the oil is very much reduced. The objection to water cooling is the danger of leakages of water contaminating the lubricating oil with most disastrous results to the main bearing surfaces. A very small quantity of water mixing with lubricating oil greatly reduces its lubricating efficiency.

On the other hand, lubricating oil is a high priced commodity, and with all Diesel engines the consumption is relatively high, especially in comparison with turbine machinery. Using lubricating oil for the purpose of cooling the pistons tends to lessen the useful life of the oil, and if the supply to the piston crowns should be intermittent carbonization rapidly takes place.

The crosshead adopted on all Tosi engines is of unusual design. With all other designs of Diesel engines the type used has two top end bearings, the gudgeon pin being fixed to the piston rod. With Tosi engines the gudgeon pin is fixed in the connecting rod, and the single bearing is in the piston rod. With this type great care has to be exercised in the design of the connecting rod to ensure a sufficient section of the fork to allow for the bending moment consequent upon the piston load and especially is this the case with a connecting rod drilled for forced lubrication.

The connecting rod of the engine illustrated is relatively short, being 3.8 cranks, and to minimize the side thrust on the guides the center line of the cylinder is set slightly out of line with the center of the crank shaft, as will be seen on reference to Fig. 2. The unique feature of the Tosi cross-head, however, is the gudgeon pin, which, as usual with a single top end bearing, is pressed into the eyes of the forked connecting rod. Outside the gudgeon pin a sleeve is keyed on. The outside of this sleeve forms the bearing surface, and around this sleeve are the usual bearing bushes let into the end of the piston rod. Adjustment for wear is made by the square-headed pins provided with a locking plate, all as shown in Figs. 1 and 2. The gudgeon-pin bearing is force lubricated from the main supply through the drilled connecting rod. The radial slippers of the crosshead are attached to the piston rod and work in the guides—one for ahead and one for astern—formed in the framing as already described. Means for adjustment of the slippers or guides are provided.

With the type of construction adopted the working pressure of the top end bearing is reduced to the moderate figure for forced lubrication of 1,600 pounds per square inch without an unduly long pin being necessary, so minimizing the bending moment for which the forked end of the connecting rod has to be designed. The overall dimensions of the crosshead and slippers are such that, with the bottom end disconnected, the piston rod, crosshead and connecting rod can be withdrawn with suitable tackle through the cylinder. The usual method of dismantling these parts with Diesel engines is to withdraw from below as has often been described in these pages.

Access from below to adjust main bearings, etc., is provided through large crank chamber doors. The chief concern will be the procedure to be adopted in order to clean or renew the main piston rings which require a certain amount of attention, depending upon the quality of fuel oil being used, etc. If a piston can only be withdrawn from the top the cylinder head must be removed, necessitating the breaking of the connections belonging to the cooling-water supply, fuel, fuel injection air, air suction, exhaust, etc. Some of these are high-pressure connections and better left undisturbed. The disconnecting of the valve gear also becomes necessary. On the other hand, removal of the pistons from below often means a higher engine. The connecting-rod bottom is not of the marine type with palm end, but is as shown in Fig. 2, which design tends to reduce its size and so facilitates the consideration of withdrawal through the cylinder.

The crankshaft for this size of engine is in two pieces, a six-throw shaft for the main engine and a two-throw for the compressors, and it is conveniently made of the solid type. It is drilled for forced lubrication. The lubricating oil is delivered by the pump to the main bearings, passing along the crankshaft to the crank pins, up the connecting rod to the gudgeon-pin bearings, through the pipe in the hollow piston rod to the piston, and down outside this pipe to the crank chamber and to the sump in the bedplate from which the lubricating oil pump draws. The pressures at full piston load on the crankpin and main bearings work out at 1,100 pounds per square inch and 340 pounds per square inch respectively.

The cylinder head of cast iron contains five valves, the fuel injection valve (6), two inlet valves (7) and two exhaust valves (8). The provision of two inlet and two exhaust valves, which is exceptional for this size of engine, makes easier the provision of ample passage for the air and exhaust, and the fact of having two small exhaust valves in place of a single larger one renders unnecessary water cooling of the exhaust valves. These valves (see Fig. 4) have cast-iron heads fixed on to steel stems, the better to resist the action of the gases; the baffle to protect the valve-spindle guide from the action of the high-velocity gases will be noted. The arrangement of the valves does not follow normal practice, in that they are not housed in separate cages, but seat direct in the cylinder head. Valve removal requires the lifting of the cylinder head, and a damaged valve seat requires the fitting of a spare head. The casting of the cylinder head is, however, simplified, and the cooling of the exhaust passage by the water circulation is more direct. The valve springs are neatly housed, although inspection of the springs requires dismantling of the valve driving gear.

The inlet valves draw air from the engine room through the grids shown in Fig. 2, through the pipes (11), Fig. 1, from the underside of the working pistons and from the interior of the engine framing (10). The undersides of the pistons are completely isolated by the diaphragm and gland (see Fig. 2) from the interior of the crank chamber. This isolation, which can generally be effected with four-cycle crosshead engines, is most desirable. Gases and burnt lubricating oil mixed with a certain percentage of fuel oil passing the main piston rings would otherwise cause contamination of the lubricating oil in the crank chamber. Leak-offs are provided for this oil as shown, and in certain cases, after filtering, this oil has been used over again as fuel. The suction of the main cylinders and of the compressors being coupled up and taken from this common source, makes for silence, keeps gases from entering the engine room and tends to cool the main engine framing. The quality of air drawn into the main cylinders and compressors is impaired in this way to some extent, and although not of much importance where the main cylinders are concerned, is not a commendable feature with high-pressure compressors (1,000 pounds per square inch delivery pressure).

The nature of the joint between the exhaust manifold (9) and the cylinder heads may be seen from Fig. 2. Allowance is made for expansion, and the supporting feet of the exhaust pipe are free to slide. The exhaust pipe is water-jacketed, with inspection doors opposite each cylinder. The valve gear is of novel design for four-cycle engines, the leading considerations having been, facility of reversing and of disconnecting for removing the cylinder heads.

The camshaft (13) is level with the cylinder tops, and is driven by spiral gearing from the crankshaft. There are two cams for each set of valves, *i. e.*, one set for ahead and one for astern. Motion is transmitted from each cam to a lever, and from the lever mounted on the eccentric fulcrum shaft by means of three pull rods to the adjustable tappets on the bell cranks operating the valves. Driving out one pin in the end of each pull rod frees the cylinder head from the valve gear, and on re-assembling the gear is replaced in exactly the same relation to the crankshaft. The cams run in an oil bath (12), and forced lubrication is provided for the valve gear.

Two tandem compressor cylinders, as will be seen from Fig. 1, are driven from a two-throw crankshaft coupled to the forward end of the main six-throw shaft. These cranks are at right angles to each other. The forward lower cylinder (18) serves to compress air for auxiliary purposes to a pressure of 60 pounds to 70 pounds per square inch, and for starting the engine as later explained. The upper cylinder (17) is the high-pressure cylinder of the three-stage compressor. The lower after tandem cylinder (15) is the low-pressure cylinder of the compressor, and also a starting cylinder, while the upper (16) is the intermediate-pressure cylinder. The capacity of this high-pressure fuel-injection air compressor is such that the swept volume of the low-pressure piston is 18.75 cubic feet per hour per brake horsepower of the main engine. The compressor is provided with automatic valves, as at (22) and (23), to all the stages, and intercoolers reduce the temperature of the air between the stages of compression. Fig. 3.

The design of the driving gear, crossheads and so forth, is exactly the same as for the main engine. At the forward end of this two-throw crankshaft and driven at reduced speed of revolution by spur gearing, are the forced-lubrication oil-circulating pump, cooling-water pump, daily service fuel-oil pump, and bilge pumps suitably grouped together (26). This reduction gearing gives the pumps a speed of revolution of 70 revolutions per minute, and so reduces the shocks of operation of the water pumps.

One of the most interesting features of all marine engines is the arrangement for starting, reversing, control, etc. With the Tosi engine the main cylinders are not used for starting. The disadvantages of introducing starting air to the main cylinders to give the first impulses are considerable. The expansion of the starting air cools the cylinder and its contents, whereas the principle of starting is to revolve the engine at such a speed of revolution that the heat generated by the compression of the air will ignite the injected fuel. For this reason the compression pressure of the Diesel engine is generally governed by the starting condition, and must be such that, allowing for contingencies, sufficient heat will be generated at the relatively low speed of revolution attained when running on starting air to counteract the cooling effect of the expansion of the starting air and ignite the injected fuel. For this reason a much higher compression pressure must normally be carried than, excepting for the starting conditions, would be required. High compressions are not desirable. Again, with slightly abnormal conditions the quantity of fuel that is injected into the working cylinder before ignition takes place may be considerable, and may give rise to high pressures to deal with which safety valves are often fitted to the main cylinders. With the Tosi

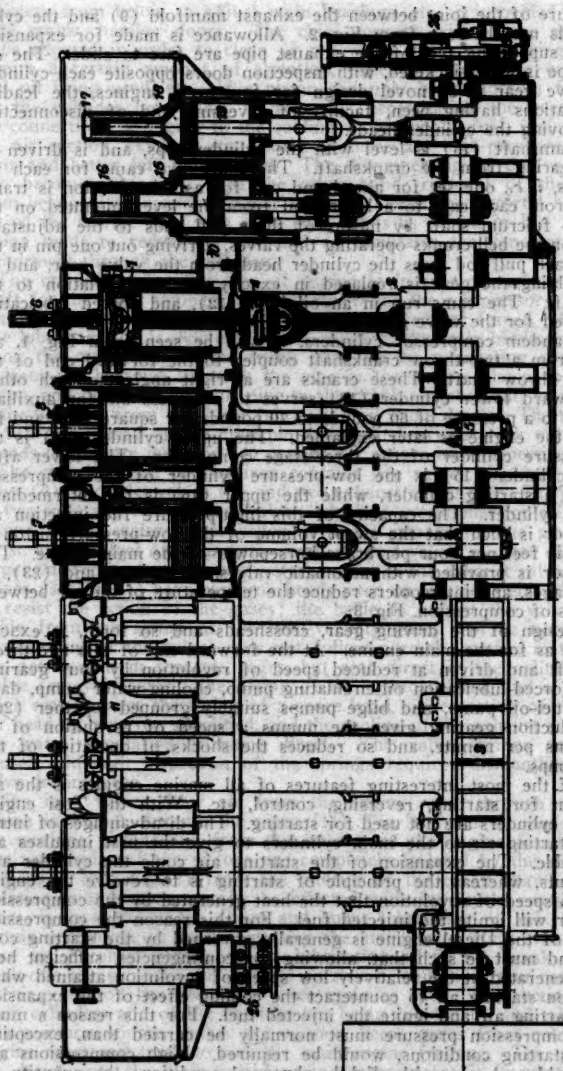


FIG. 1.

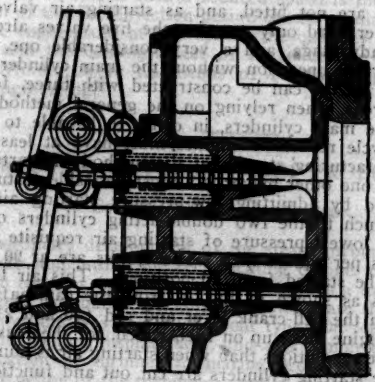


FIG. 4.

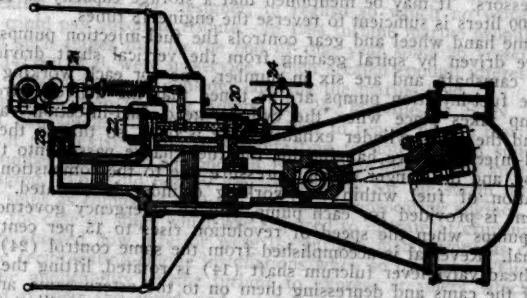


FIG. 3.

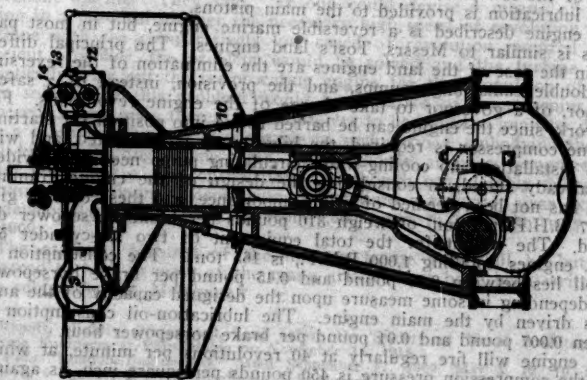


FIG. 2.

engine safety valves are not fitted, and as starting air valves are also eliminated the cylinder head only contains the five valves already enumerated. A further advantage, and a very considerable one, is that by carrying out the starting operation without the main cylinders a marine reversible four-cycle engine can be constructed with three, four, six or eight cylinders, whereas, when relying on the general method employing starting valves in the main cylinders, in order to be able to start from any position, four-cycle reversible engines must have at least six cylinders. From a manufacturing standpoint the method of starting adopted by Tosi and used by one other maker is a most desirable feature.

Starting is effected by admitting low-pressure starting air some 200 pounds per square inch to the two double-acting cylinders of the compressor group. The lowest pressure of starting air requisite to start the engine is 170 pounds per square inch. The cranks are at 90 degrees, so that the engine can be started from all positions. This air is controlled by the piston valves as shown in Fig. 3, from the camshaft driven by special cams through the bell-crank lever and rod (20 and 21).

Immediately the engine picks up on fuel, which, for the reasons already stated, requires fewer revolutions than when starting air is admitted to the main cylinders, these starting cylinders are cut out and function normally as compressors. It may be mentioned that a storage capacity of starting air of 1,000 liters is sufficient to reverse the engine 18 times.

The same hand wheel and gear controls the fuel-injection pumps (25), which are driven by spiral gearing from the vertical shaft driving the overhead camshaft, and are six in number, one for each working cylinder. The fuel-injection pumps are so timed that the delivery stroke of each pump takes place when the cylinder-head fuel-injection valve is closed and the main cylinder exhaust valve is open, so that in the event of a fuel-injection valve hanging up, the fuel charge passes into the exhaust pipe, and the danger of high pressure due to the combustion of an accumulation of fuel within the working cylinder is obviated. Hand adjustment is provided for each pump, and an emergency governor cuts out the pumps when the speed of revolution rises to 15 per cent above the normal. Reversal is accomplished from the same control (24). The cylinder-head valve-lever fulcrum shaft (14) is rotated, lifting the ahead levers off the cams and depressing them on to the astern cams, and also altering the setting as required of the piston valves controlling the entrance of the low-pressure air to the two starting cylinders. Sight-feed forced lubrication is provided to the main pistons.

The engine described is a reversible marine engine, but in most particulars is similar to Messrs. Tosi's land engines. The principal differences in the case of the land engines are the elimination of the reversing gear, double cams, bilge pumps, and the provision, instead of a safety governor, of a governor to take charge of the engine revolutions. For land work, since the engine can be barred round into position for starting, only one compressor is required, the bilge pump is eliminated, and with certain installations no cooling-water circulating pump need be provided.

As already stated, in considering the design of the engine framing, weight has not been deemed of prime importance, and these engines give some 7 B.H.P. per ton, or weigh 310 pounds per brake horsepower developed. The weight for the total equipment of two six-cylinder 500 B.H.P. engines, totalling 1,000 B.H.P., is 165 tons. The consumption of fuel oil lies between 0.42 pound and 0.45 pound per brake horsepower hour, depending in some measure upon the designed capacity of the auxiliaries driven by the main engine. The lubrication-oil consumption is between 0.007 pound and 0.01 pound per brake horsepower hour.

The engine will fire regularly at 40 revolutions per minute, at which speed the compression pressure is 450 pounds per square inch, as against 485 pounds per square inch at 165 revolutions per minute. One reason

for the capacity to run at such a reduced speed is the fact that the phase and duration of the fuel-injection period can be regulated according to the speed of the engine, and at low speeds the quantity of fuel-injection air is reduced and the cooling effect of the excessive quantity that would enter the cylinder, were no means of regulation provided, is entirely absent.—"Engineering."

## PROGRESS IN TURBINE SHIP PROPULSION.*

By Mr. FRANCIS HODGKINSON.

The extraordinary development of the steam turbine for land purposes, almost entirely supplanting the reciprocating engine, rendered the application of steam turbines for the propulsion of ships a natural sequence.

The *Turbinia* was built especially to demonstrate the possibility of turbine propulsion and was of torpedo-boat type, 100 feet long, 9 feet beam, 3 feet draught, 44 tons displacement, and was completed in 1895.

The application of turbines to marine propulsion became very rapid, and the construction of the turbine as applied to the larger ships called for the greatest skill of the designer, in regard to which no words of the author could pay adequate tribute. In 1904 turbine marine propulsion had reached such stages that by the end of that year some 26 vessels, aggregating 147,000 H.P. were in service, which created enough interest to lead George Westinghouse to consider entering the marine field.

At his instigation the late Admiral G. W. Melville, U. S. N., and Mr. John H. Macalpine, visited Europe in 1904 to report on the marine situation as it then existed, and it is interesting to note that, in spite of the large number of vessels equipped in Europe, the report was anything but encouraging. They did not see any opportunity for the steam turbine for driving ships as compared with the existing reciprocating engines, particularly in the case of ships which are required to operate at cruising speeds, and they showed how inferior were the turbine installations as compared with the reciprocating-engine installations when so operated. They recognized the applicability of the turbine to express steamers which always operated at their full speed, and pointed out their lack of economy when applied to lower-speed ships which comprise the greater part of the merchant marine. They said "If one could devise a means of reconciling in a practical manner the necessary high speed of revolution of the turbine with the comparatively low speed of revolution required by an efficient propeller, the problem would be solved and the turbine would practically wipe out the reciprocating engine for the propulsion of ships. The solution of this problem would be a stroke of great genius," and proceeded some time thereafter themselves to become the genii by designing a tooth reduction gear adaptable to such service, embodying the now well-known floating frame. Mr. Westinghouse at once recognized the great possibility of its construction, and though considerably delayed on account of the financial depression at that time, a reduction gear expected to develop 6,000 H.P., reducing from 1,500 r.p.m. to 300 r.p.m. was tested in 1909.

The essential features of the floating frame gear will be but little referred to here in view of the many articles that have been written on this subject by Mr. McAlpine.†

*Report, slightly abbreviated, read before the Society of Naval Architects and Marine Engineers at Philadelphia on November 14, 1918.

†*Vide* "London Engineering" dated September 17, 1909, May 6, 13, 20, 27 and June 3, 9 and 16, 1916; Proceedings of Engineers' Society of Western Pennsylvania dated December 18, 1917; Institution of Naval Architects, London, March 29, 1917.

The point most freely discussed in connection with reduction gears is the permissible tooth pressure per lineal inch of tooth face. Unquestionably the safe load which can be transmitted by the gear teeth is all-important, but unfortunately the subject has been frequently regarded as entirely independent of any other feature of the reduction-gear design. This is a mistaken view. The ultimate limit of pressure which the teeth can safely carry has not been determined, but the pressures at present in use among designers are only possible with the most accurate alignment. No comparison of tooth pressure is therefore reliable without careful consideration of the methods used to maintain the unit pressure within the limits for which the design was made. Faulty bearing design, inadequate support or improper couplings may be individually or collectively responsible for deranging the alignment, resulting in a concentration of the tooth load and often failure of the teeth. It is obvious, then, that merely to limit the unit tooth pressure to some fixed empirical figure for all gears in no way insures the safety of the teeth. On the other hand, anything which tends to maintain the alignment of the teeth and to prevent concentration of pressure is to be welcomed as a source of safety in the operation of the gearing. It is for this reason that the Melville-Macalpine invention of the I-beam support for the pinion bearings is so thoroughly justifying the claims made for it. The closer this mechanically-correct device is investigated, the more apparent becomes its most important function, namely, to maintain automatically a very uniform distribution of tooth pressure under the most severe distortions of the gear-case that can occur in the frailest hull. Its value in permitting higher unit loads than are customary on which have been termed "rigid bearing" gears is graphically illustrated in Fig. 1, subjoined. The curves of the rigid-bearing dimensions have been com-

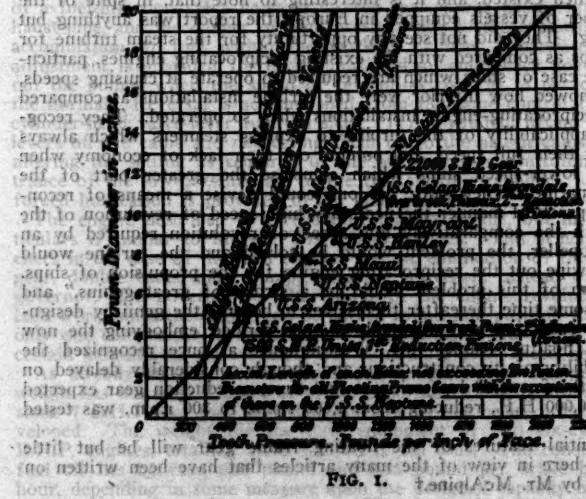



FIG. 1.

piled from recent marine practice in Great Britain. The floating-frame figures are those used by the Westinghouse Electric and Manufacturing Company for marine work, and there are points indicated on this curve showing gears actually in service.

As previously referred to, the gear by permitting high turbine speed, much simplifies the turbine design. It enables a speed to be selected for the turbine appropriate for best economy. Furthermore, it inherently permits a design of increased reliability by virtue of sensibly reduced dimensions. The small high-speed geared turbines require few of the elaborate precautions in warming up and getting in service that must be practiced with the large direct-connected turbines.

Gears may readily be arranged with two pinions for even the smallest single-screw steamer, permitting the employment of what are known as cross-compound turbines; that is, a high-pressure and a low-pressure turbine, the steam passing them in series. Piping is so arranged that by the manipulation of valves the high-pressure turbine may exhaust direct to the condenser, or the high-pressure turbine be isolated and the low-pressure turbine receive high-pressure steam direct. Thus, generally speaking, no turbine or gear accident is likely to so incapacitate the machinery that the vessel cannot reach port at reduced speeds. There is one record of a 10½ knot steamer, which, on account of breakage of one pinion, crossed the Atlantic with the low-pressure turbine alone, obtaining speeds as high as 9 knots.

The general arrangement of such an installation is shown in Fig. 2,

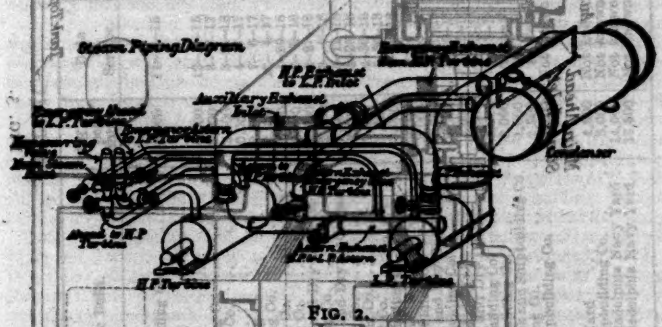



FIG. 2.

annexed. Its reliability has much to commend it. This is not only due to the flexibility above referred to, but further due to the simplification of the turbines themselves. Each turbine only expands the steam corresponding to half the heat drop, and so has a much reduced number of turbine elements, smaller temperature range, and reduced length between bearings as compared with a complete expansion turbine, having similar economy. Both high and low-pressure turbines are provided with reversing elements arranged in a similarly compounded manner to the ahead elements, thus rendering maneuvering possibilities as complete with one turbine element operating as with both.

Of necessity the cross-compound turbine installation with its double-pinion gear is more costly than the single complete-expansion turbine, and with keen competition will not be installed unless shipowners recognize its manifest advantages. So far as the writer has observed, owners seem quite indifferent to the probable reliability of machinery so long as they secure insurance and the protection of the classification societies. The plea is therefore made by the author that underwriters, in collaboration with the classification societies, should discriminate in insurance rates between types of installations having manifestly differing degrees of reliability.

No doubt single-screw steamers will frequently be driven by single complete-expansion turbines, at least for low powers. An example of a small-sized installation, 1,500 H.P., is shown in Fig. 3, and attention is

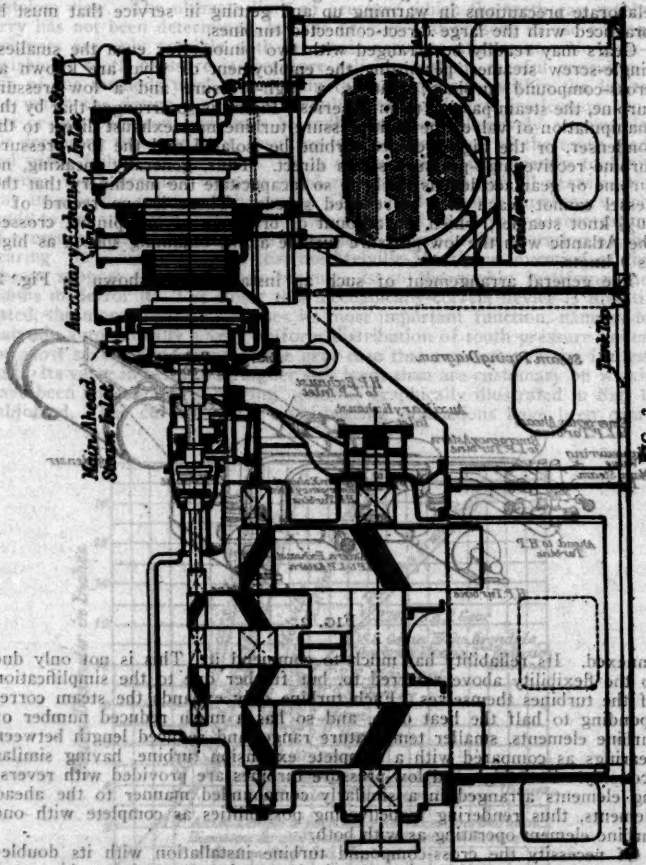



FIG. 3.

called to some of its features. The double reduction-gear elements are arranged one above the other with the high-speed pinion uppermost. This elevates the turbine axis and permits the condenser to be located immediately below the turbine, permitting the exhaust to issue from the underside of the turbine and pass vertically downwards to the condenser, much simplifying the removal of water. One end of the turbine is carried directly upon a bracket integral with the gear case, the other by the

TABLE II.

| Name of Ship. | Where Built. | S.H.P. per Ship. | Date in Service. | Miles Travelled (approx.). | Troubles in Service. | Remarks. |
|---|---|---|---|---|---|---|
| Melville | New York Shipbuilding Co. | 4,000 | 11- 1-15 | 7,000 | None | U.S. T.B.D., tender main gear only. |
| Neptune | Maryland Steel Co. | 8,500 | 12- 1-15 | 128,000 | See Note A | U.S. collier. |
| Malmatger | Chester Shipbuilding Co. | 2,900 | 5- 8-17 | 2,500 | None | Sunk 9-23-17. |
| Westwood | Ames Shipbuilding Co. | 2,500 | 9-24-18 | 16,000 | None | |
| Mazi | Union Iron Works | 10,000 | 6-17-17 | 60,000 | See Note B | |
| Golaa | Chester Shipbuilding Co. | 2,900 | 7- 6-17 | 90,000 | See Note C | |
| Hisko | Chester Shipbuilding Co. | 2,900 | 12- 4-17 | 25,000 | None | |
| Odronado | Moore Shipbuilding Co. | 2,400 | 1-28-18 | 27,000 | None | |
| Yosemite | Moore Shipbuilding Co. | 2,400 | 2- 2-18 | 27,000 | None | |
| Westerley | Ames Shipbuilding and Dry Dock Co. | 2,500 | 5-27-18 | 7,800 | None | Sunk 4-24-18. |
| Sudbury | Chester Shipbuilding Co. | 2,900 | 5- 5-18 | 15,000 | See Note D | |
| West Eagle | Ames Shipbuilding and Dry Dock Co. | 2,500 | 4-13-18 | 11,000 | None | |
| Yellowstone | Moore Shipbuilding Co. | 2,400 | 4-22-18 | 22,000 | None | |
| Overbrook | Chester Shipbuilding Co. | 2,900 | 4-25-18 | 16,000 | None | |
| Oakland | Moore Shipbuilding Co. | 2,750 | 4-25-18 | 19,000 | None | |
| Ascomoc | Los Angeles Shipbuilding Co. | 3,000 | 4- 3-18 | 12,000 | None | |
| Avondale | Chester Shipbuilding Co. | 2,900 | 4-23-18 | 6,000 | See Note E | |
| Wakulla | Los Angeles Shipbuilding Co. | 3,000 | 4-25-18 | — | None | |
| West Ford | Ames Shipbuilding and Dry Dock Co. | 2,750 | 4-30-18 | 8,000 | See Note F | |
| Wampum | Los Angeles Shipbuilding Co. | 3,000 | 5- 9-18 | — | None | |
| Wassaw | Los Angeles Shipbuilding Co. | 3,000 | 7-25-18 | — | None | |
| Montrolfie | Ames Shipbuilding and Dry Dock Co. | 2,500 | 4-13-18 | 5,000 | None | |
| West Geloc | Los Angeles Shipbuilding Co. | 3,000 | 5-13-18 | — | None | |
| West Gleata | Los Angeles Shipbuilding Co. | 3,000 | 5- 5-18 | — | None | |
| Polar Sea | Baltimore Dry Dock and Shipbuilding Co. | 1,500 | 5- 8-18 | — | None | |
| Phomix | Chester Shipbuilding Co. | 2,900 | 6- 7-18 | — | None | |
| Pennsylvania | Newport News Shipbuilding Co. | 8,500 | Not known | Not known | None | U.S. battleship cruising gears only. |
| Arizona | New York Navy Yard | 4,000 | Not known | Not known | See Note G | " " |
| Mississippi | Newport News Shipbuilding Co. | 4,000 | Not known | Not known | None | " " |
| Henley | Re-engined at Philadelphia Navy Yard | 14,000 | Not known | Not known | None | U.S. T.B.D. |
| Mayrant | Re-engined at Philadelphia Navy Yard | 13,500 | Not known | Not known | None | U.S. T.B.D. |

exhaust outlet being set directly upon the condenser. The condenser is secured by its upper surface to seatings, eliminating disturbances to alignment on account of expansion of the condenser.

There appears to be much apprehension in the matter of the reliability of turbine-geared steamers, and if current rumor may be relied upon there have been many failures, which, with more careful design or by employing the best in the art, would seem to have been wholly unnecessary. To correct such an impression, Table II gives a list of all vessels equipped with the floating-frame gear and Westinghouse turbines. The table states the date on which the vessel entered service, the power of the installation, and approximately the miles traveled up to the present time (September 1, 1918).

Some troubles have been experienced as are certain to occur with all kinds of machinery. Since there must be a large human element, footnotes are appended, plainly stating the character of the troubles.

*Note A.*—U. S. S. *Neptune.*—This ship was equipped with two sets (twin screws) single-reduction gears and turbines, the turbines having been largely experimental and were not successful. The gears, however, operated splendidly. The turbine machinery was then replaced, and inasmuch as a higher turbine speed was considered desirable, new gears involving a greater speed reduction were also installed at this time. The designed constants of the new gears were very high and are still much in advance of our present practice.

Considerable trouble was experienced due to the teeth abrading which was only overcome by the use of lard oil. Since that time, however, in November, 1915, no further trouble has been recorded, and the teeth at the last inspection were in good condition without any indication of wear.

*Note B.*—S. S. *Manx.*—This vessel was built at the Union Iron Works. Inasmuch as we were late in the delivering the machinery, we were unable to carry out any tests on the gears in the works, and inasmuch as great store was set by the owners that nothing should go wrong on the maiden voyage, and as there was some uncertainty as to the excellence of the tooth surfaces, lard oil was employed as a lubricant for the first two voyages, after which mineral oil was substituted for the lard oil.

The operation of the machinery was all that could be desired for eight voyages, at which time there was abrasion of the gears due to failure of the lubrication. This was discovered in Honolulu and the ship returned to San Francisco with nothing being done to the machinery except an attempt to clean the oiling system. The tooth surfaces smoothed up considerably during the voyage.

The gears were then transposed from one side to the other so as to make what had been the astern surfaces the ahead surfaces, so as to use uninjured tooth surfaces for ahead operation.

Again because of some uncertainty of the tooth surfaces lard oil was employed as a lubricant and the vessel entered in Government service, going to Chile for cargo and proceeding to Baltimore. During the voyage there was an accident to an oil cooler, admitting salt water to the system. The shipbuilder having provided no means of removing any water, the effect of the salt water on the lard oil was to turn it violently acid, which necessitated a complete overhauling of the machinery on its arrival in Baltimore.

On this being completed and on the trip from Baltimore to New York trouble was experienced with abrasion of the tooth surfaces due, it was believed, to an inferior quality of oil, together with further failure of the oil cooler, admitting enough water to the system to make an emulsion of approximately 30 per cent. A better oil was substituted and nothing was done to the tooth surfaces, the vessel having made one voyage to Europe and back with no trouble as far as the gears are concerned.

During this voyage trouble was experienced with one of the turbines, causing quite an extensive repair because of an obstruction in one of the oil passages to the thrust bearing of the turbine, causing the thrust bearing to fail; the turbine rotor moving endwise, completely wrecking the labyrinth packing between the ahead and astern elements and buckling the rotor.

*Note C.*—S. S. *Golsa.*—Norwegian oil tanker built by Chester Shipbuilding Company, and chartered by British Admiralty. Put in service July 6, 1917. Made three round trips to English ports. At present in coastwise trade between Port Arthur, Texas and Philadelphia or Bayonne. Towards end of last trip from Europe (February, 1918) some teeth on starboard high-speed pinion broke out. This gear was disconnected and ship made port, using low-pressure turbine only. Inspection showed break occurred on one helix, 3 inches from end of tooth face. There was no evidence of wear on any of the tooth faces, the undamaged portion of broken tooth face indicating uniform distribution of load. This fact and the position of fracture make certain accident was due to defective material.

In August of this year the high-pressure turbine suffered injury as a result of improper setting of the turbine thrust bearing. Shortly thereafter an accident occurred to the pinion which replaced the one originally defective. The character of the broken tooth was similar to the previous one. The broken parts have not yet been received to enable a careful examination of the material to be made.

*Note D.*—S. S. *Sudbury.*—An American freighter built by the Chester Shipbuilding Company for the Shawmut Steamship Company. It is at present being operated by the United States Naval Overseas Transportation Service. A few days after leaving France on return half of maiden voyage, some teeth broke out of starboard high-speed pinion. This gear was disconnected and ship returned to New York using low-pressure turbine only, with which 9 knots was attained, the designed full speed being 10½ knots.

On arrival at port, inspection showed no wear, the tooth load having been well distributed on all tooth faces. Several teeth were cracked. Subsequent examination of steel indicates defective material, most probably due to faulty heat treatment, the fracture indicating the steel had been burned. This was confirmed by discovery that some teeth in port high-speed pinion were also cracked, although no failure occurred in service. Both pinions were forged from the same billet and treated at the same times. These pinions are similar as to material and design with 25 other pinions now in successful operation.

*Note E.*—S. S. *Avondale.*—An American oil tanker, built by the Chester Shipbuilding Company, and being operated by the Pan-American Oil Company for the United States Shipping Board. Just as ship was leaving Philadelphia for trial trip a heavy blade rub was heard in high-pressure turbine. Inspection indicated defective workmanship. Some of the rotating blades were loose as a result of having been improperly put in. The spindle was rebladed and ship proceeded satisfactorily from Philadelphia to New York. This turbine is a counterpart of 12 others now in successful operation on various ships.

*Note F.*—S. S. *West Ford.*—This vessel left Seattle at 5 P. M. on July 14, 1918, on her maiden voyage to an Atlantic port. Seven hours out from Seattle, oil commenced foaming out from the turbine, running over the engine-room gratings. The chief engineer was called and, instead of investigating the reason for augmentation of the oil, he was content to pump 200 gallons to a reserve tank. Later the same thing occurred, oil again running out on the gratings, and later still at 2.14 A. M., the high-pressure turbine commenced vibrating so badly that it interfered with their lighting system. They shut down to remedy the matter of the lighting

system, and on trying to start the turbine it required 125 pounds initial pressure to revolve it. Serious vibration was again in evidence, so the chief decided to do what he termed "crank up on the thrust." It was impracticable to anchor, so they returned to Port Townsend at 40 r.p.m. They then made an observation of the oil in the drain tank, and on removing the cap the oil squirted to the roof of the shaft alley. It was later discovered an oil cooler tube had split, admitting large quantities of salt water to the oil system.

Instructions had been given that the level of the oil in the drain tank should be regularly observed, and it is incomprehensible that it was not at once obvious that something was getting in the oil system from some exterior source. Merely looking at the oil would have determined this was salt water. Then all that would have been necessary was to shut off the water service to the cooler and separate the relatively small quantity of salt water from the gravity tank, and there need have been no interruption whatsoever. The result of such neglect was for the bearings, for want of lubrication, to let the rotor down, causing considerable injury to the turbine.

*Note G.*—U. S. S. *Arizona.*—On a voyage from Cuba to the United States the teeth of one of the cruising gears scored. Subsequent investigation showed that the oil pumps had stopped and the supply to the gears had failed. The teeth were dressed up and no further trouble has been experienced.

As in land installations, the overall economic performance is not dependent on the prime mover alone. High performance is not to be secured without care being bestowed on all the auxiliary apparatus, attention to matters of heat balance with the feed-water heater, condensing apparatus capable of performing with a close approach to the ideal, &c.

Large size land turbines have been built which deliver to the shaft not less than about 80 per cent of the theoretical energy available from the steam expanding between the limits specified, so that further improvements in the turbine itself will not materially raise this efficiency. Further improvement in economy must be looked for from causes other than the prime mover itself. This is a subject of the greatest importance in view of the rapidly increasing cost of fuel, and it justifies considerably more capital expenditure for economizers and other such apparatus which will reduce fuel cost.

The engineer of our large central stations is subject to keen competition and is confronted almost solely with the problem of producing reliable energy for the least cost, including fixed charges. He has before him daily statements showing his overall power costs with a high degree of accuracy, resulting in considerable success on his part in the production of low power costs.

It is therefore not unreasonable that the marine engineer might look to the central-station engineer for advancement from some of the latter's best practice. It is, of course, understood that complete standby machinery is not available for the marine engineer, neither can the superintending engineer direct those in immediate charge of the machinery by frequent visits. Hence there is the more urgent call for the greatest reliability and simplicity of the marine installation; though, after all, perhaps not more than that which the central stations have in our big cities. An interruption to service, however brief, is regarded most seriously.

*Feed-Water Heating Systems.*—It is common practice on shipboard to provide a closed or tubular heater to which is led the exhaust steam from the auxiliary machinery, which latter exhausts at a pressure between 5 pounds and 10 pounds gage. The exhaust system is provided with a connection to let any proportion of this steam to the condenser. Sometimes again there is a hand valve, permitting any surplus of this steam to be

taken to a low-pressure stage of the turbine, all of which calls for hand manipulation where the conditions are sometimes quite variable, owing to more or less intermittent operation of the feed pump. Sometimes the connection to the condenser is provided with a spring-loaded valve so that when the exhaust pressure reaches a predetermined pressure, higher than that within the condenser, steam may pass to the condenser. The construction of the valves usually employed is such that the higher the pressure in the condenser, the higher is the pressure at which the steam will pass to the condenser, which is, of course, undesirable. A system which will automatically maintain a predetermined temperature in the feed heater at all times, bleeding the main turbine for this purpose, at high powers if necessary, and at the same time automatically permit any surplus steam not condensed in the feed-water heater to do useful work in the turbine, is much to be desired.

Heat balance systems have for some time been available for land purposes, there being provided automatic valves so that whenever there is a surplus of auxiliary exhaust steam this surplus may pass through a low-pressure stage of the turbine, doing work therein at the rate of 20 pounds to 22 pounds per horsepower. Other types of valves are available for land installations which will not only take surplus steam into the turbine at times of light loads on the main unit at which time the auxiliaries will obviously furnish too much exhaust steam, but, on the other hand, they are arranged to deliver steam from the turbine to the feed-water heater at times of heavy loads when the stage pressures in the turbine are relatively high and there is a deficiency of steam for feed-water heating. The latter should particularly find a place on war vessels which are

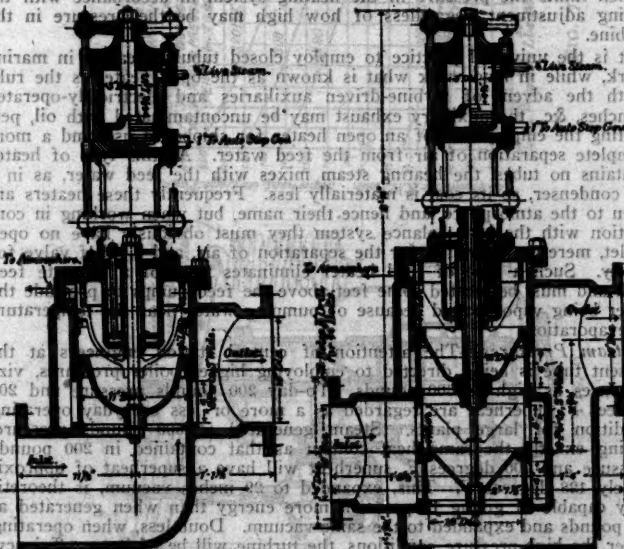


FIG. 4. FIG. 5.

required to operate for long periods at different speeds, where there is a widely varying ratio of steam used by the auxiliary machinery to that used by the main turbines. In merchant vessels this ratio is more nearly constant, but nevertheless careful study to the end of obtaining a complete heat balance will be advantageous. Details of the two forms of valves above described are shown in Fig. 4 and Fig. 5, subjoined.

The valve shown in Fig. 4 is intended for cases where there is never a deficiency of auxiliary exhaust steam for feed heating. It comprises a piston having atmospheric pressure above it, and the auxiliary exhaust steam below. When the auxiliary exhaust steam pressure rises a sufficient amount above the pressure of the atmosphere to raise the piston, steam may pass to the turbine. Air leakage into the turbine is entirely precluded by ample steam seal around the piston furnished by the auxiliary exhaust steam. The piston may be loaded to any feed-heater pressure desired. The upper cylinder is to forcibly close the valve in the event of the turbine over speeding.

The modification of this valve shown in Fig. 5 is adaptable where there are at times a deficiency of auxiliary exhaust steam. Any time the stage pressure in the turbine is greater than that in the heater the lower piston will raise, permitting passage of steam from the turbine to the heater at the same time, when required, permitting flow to the turbine as in the case of the valve shown in Fig. 4.

It sometimes occurs that the turbine is subjected to such overloads that a stage of the turbine selected to give the desired heater pressure at moderate loads would be far too high for the heater at overloads. In this instance the modification shown in Fig. 6, subjoined, may be employed which limits the pressure in the heating system in accordance with the spring adjustment, regardless of how high may be the pressure in the turbine.

It is the universal practice to employ closed tubular heaters in marine work, while in land work what is known as the open heater is the rule. With the advent of turbine-driven auxiliaries and electrically-operated winches, &c., the auxiliary exhaust may be uncontaminated with oil, permitting the employment of an open heater for shipboard use, and a more complete separation of air from the feed water. As this type of heater contains no tubes, the heating steam mixes with the feed water, as in a jet condenser, so its cost is materially less. Frequently these heaters are open to the atmosphere, and hence their name, but when working in connection with the heat balance system they must obviously have no open outlet, merely a provision for the separation of air, and a relief valve for safety. Such a type of feed heater eliminates need of a separate feed tank and must be located some feet above the feed pumps to preclude the latter being vapor bound because of pumping water near the temperature of evaporation.

*Steam Pressures.*—The attention of central station engineers at the present time is being directed to employing higher boiler pressures, viz., pressures as high as 600 pounds. To-day 200 pounds pressure and 200 degrees F. superheat are regarded as a more or less everyday operating condition for large plants. Steam generated at 600 pounds pressure, having exactly the same heat content as that contained in 200 pounds pressure and 200 degrees F. superheat will have a superheat of approximately 128 degrees F. This, expanded to 29 inches vacuum, is theoretically capable of giving 12 per cent more energy than when generated at 200 pounds and expanded to the same vacuum. Doubtless, when operating under the high-pressure conditions, the turbine will be of lower efficiency. The high-pressure element will be less efficient on account of the great density and the small volume of the steam, and on the other hand the low-pressure elements will be less efficient because of the great amount of

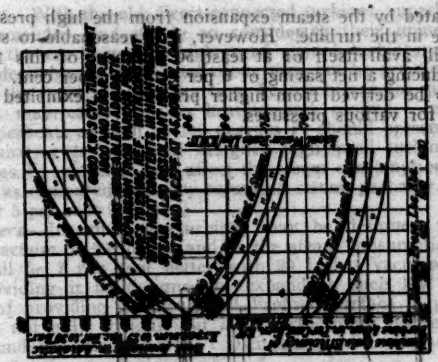


FIG. 8.

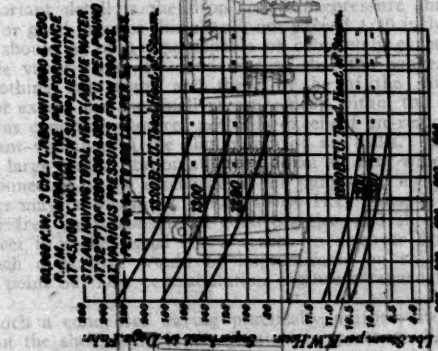


FIG. 7.

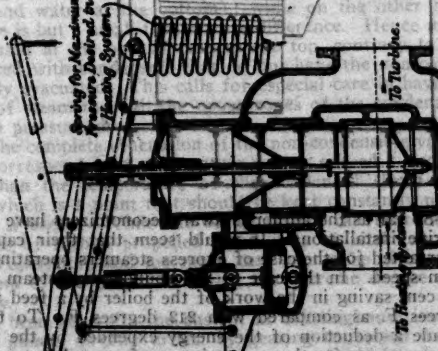


FIG. 6.

water precipitated by the steam expansion from the high pressure, introducing a brake in the turbine. However, it is reasonable to suppose that the turbine will avail itself of at least 50 per cent of this 13 per cent possibilty, producing a net saving of 6 per cent or 7 per cent. What may be expected to be derived from higher pressures is exhibited in Figs. 7 and 8, plotted for various pressures.

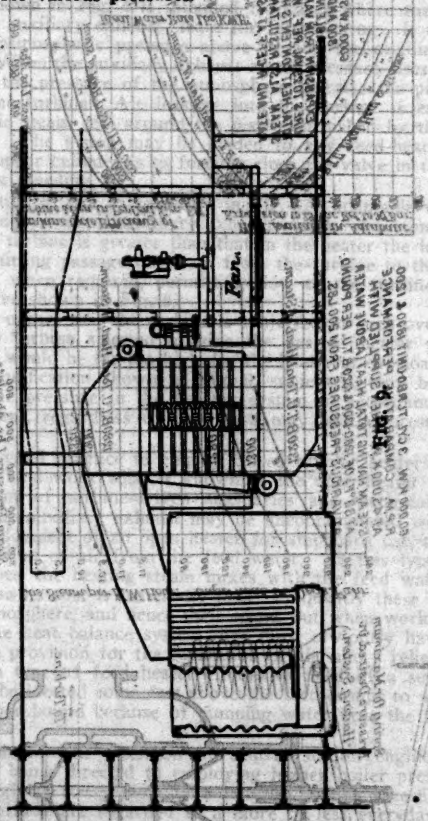


FIG. 9.

*Economizers.*—So far as the author is aware, economizers have not been employed in marine installations. It would seem that their capital cost would be well warranted for the case of express steamers operating always at their maximum speed. In the case of 250 pounds gage steam pressure there is a 16 per cent saving in the work of the boiler by a feed temperature of 375 degrees F. as compared with 212 degrees F. To this gain there must be made a deduction of the energy expended by the induced-draft fans and the scrapers for the economizer surfaces. A net gain of 10 per cent should be readily realized. Their installation would seem to present no particular difficulty as is shown in Fig. 9.

*Superheat.*—With the old direct-connected turbines, superheat has never been recommended, and wisely so considering their immense structure. Even with the much smaller high-speed geared turbines as high superheat as is desirable in land installations may not be practicable at sea on account of the rapid stopping, starting, &c., necessary for quick maneuvering, which may cause uneven heating and cooling. In the author's opinion superheats as high as 100 degrees F. should present no operating difficulty whatsoever with well designed high-speed turbines. One hundred degrees F. superheat will affect the steam consumption not less than 8 per cent as compared with dry saturated steam, probably saving some 4 per cent or 5 per cent in fuel.

*Condensers.*—Inasmuch as turbines may be designed to expand the steam to vacuum of 29 inches, and as the steam consumption with such a turbine will be 7 per cent better with 29 inches vacuum than with 28 inches providing it be designed to expand to such low pressures, the necessity of providing sufficient well-disposed surface to produce such a vacuum is not only obvious, but it is furthermore necessary to have a complete understanding of what constitutes good condenser performance with the apparatus available.

An important detail is the drop in steam pressure through the condenser. For good practice this should not exceed 1/10 inch mercury. The air pump should be of a type capable of completely evacuating the non-condensible vapor from the condenser so that the condenser shell shall contain nothing but steam, and therefore be of the same temperature throughout except for the small pressure drop within the shell itself. To produce this condition requires not only excellent air-removing apparatus, but constant vigilance in the elimination of air leaks. It is the practice of certain large central stations to provide an air bell, which may on occasion be connected to the discharge of air pumps, by means of which the air leakage may be measured. With 40,000 H.P. units, 3½ cubic feet per minute of free air is considered good practice, and should this reach to 7 cubic feet it is the signal for a search to discover the source of the leak. Such a practice would be impracticable on shipboard and only serves to point out how central-station engineers regard the ill-effects of air leaks.

With such a condenser having practically uniform steam temperature throughout the shell, there is the greatest temperature difference between circulating water and steam in the bottom pass (assuming steam enters at the top and water at the bottom); while on the other hand, in the top pass there is but a small temperature difference. Hence much more steam is condensed at the bottom than at the top, contrary to what is usually experienced with condensers that do not have the non-condensible vapors completely evacuated. This calls for especial care in having paths for the passage of steam through the upper zones of the condenser to insure the minimum pressure drop.

With the complete evacuation of the non-condensible vapors, the vacuum should correspond to a temperature not to exceed 1 degree or 2 degrees higher than the temperature of the condensed steam leaving the condenser, which is a point that should be kept constantly under observation.

*Oiling Systems.*—While the oiling system can bear no relation to the economics of turbine-propelled ships, it is of vital concern to the reliability of operation, and too much care cannot be given to the lay-out of the system to the end of obtaining the extreme degree of simplicity and reliability.

It has been generally the custom in Westinghouse installations to provide oil pumps directly driven by gearing connected to the main gears. The double-pinion double-reduction gears, of which a large number have been built, have two oil pumps, one driven from each intermediate gear

shaft. The design contemplated for the sake of reliability that each oil pump alone be large enough for the system. These pumps are relayed by an auxiliary steam-driven pump for use when maneuvering. The direct-driven pumps have been objected to in some quarters and duplicate steam-driven pumps substituted therefor. The advantage of the latter, while no doubt lacking the extreme reliability of the direct-driven pumps, is that, when properly controlled, they only pump the oil needed by the system instead of, say, three times as much, which no doubt will have much to do with increasing the life of the oil as well as simplifying the filtering or straining systems. What is regarded as an ideal system is shown in Fig. 10. It has the following features:

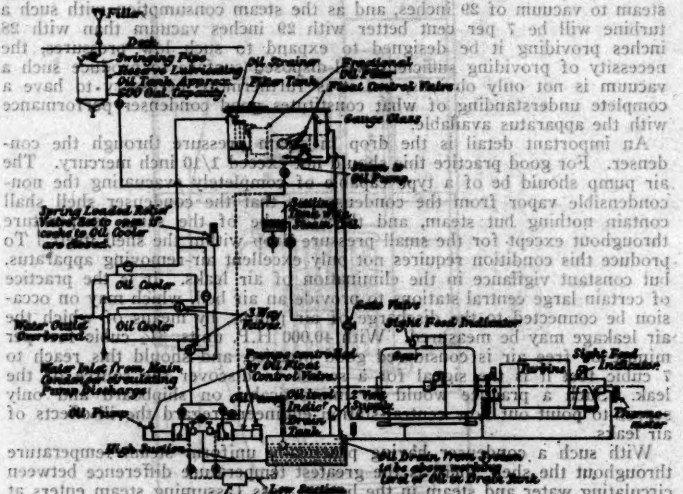



FIG. 10.

1. The pumps are located at as low a level as possible to reduce the suction head to the extreme limit.

2. The drain tank is of as large a dimension as practicable, and is located sufficiently low to insure the oil draining freely to it from the machinery with ample-sized pipes, and assurance that with the extreme rolling and pitching there will be no leakage from the bearings.

3. The pump suction should be several inches above the bottom of the tank to avoid drawing dirt and emulsion from the bottom.

4. It is regarded as desirable to employ an additional suction from the bottom of the tank by means of which dirty oil may be pumped through a filter and to permit complete emptying of the tank for cleaning purposes.

5. The pump is arranged to discharge its oil through coolers to an overhead gravity tank located some 20 feet above the machinery.

6. Duplicate coolers are employed so that while one set is in use the other may be cut out of service. They should be by-passed with a spring-loaded check valve, loaded to some amount greater than the resistance of the cooler, so that under no conditions can mishandling of the valves cause interruption to service.

7. An overhead gravity tank is provided with three strainers having successively finer mesh, the first being sheet metal with 1/16 inch perforations, the second 30-mesh wire gauze, the last 70-mesh wire gauze.

In the event of these becoming choked there can be no interruption to service, for the oil mostly flows over the top, when they may be readily removed for cleaning. The screens are secured to quite large cast frames arranged so that any dirt falling from the screen on their being removed will be caught in the frames and thus removed.

From these the oil passes through a fractional filter, that is, some of the oil runs through filter bags and is thoroughly filtered, the rest overflowing. The gravity tank should be provided with a very rugged float of large power for controlling the speed of the oil pumps. The outlet from the tank to the machinery should be well above the bottom and there should be other means of drawing oil from the bottom of this tank to a filter or settling tank. The character of the oil in the bottom of this gravity tank should be regularly observed to be sure that the water is not entering the system from any source.

8. The system which provides a settling tank capable of taking a complete charge of oil for the system is to be recommended.

9. It is considered desirable that the oil coolers should be arranged, if possible, to have the oil pressure superior to the water pressure, so that a loss of oil will occur rather than the admixture of salt water. The former would be shown by the oil level indicator which is situated in the lower tank.

There are installations, such as destroyers, where there is but little head room and where a gravity-oiling system is impracticable and a pressure-oiling system must be employed. In this case twin strainers or their equivalent must be resorted to. These are, of course, dangerous because of them becoming choked and interrupting the service, or bursting and letting dirt through if the pumps are sufficiently powerful. If they must be employed they should invariably be by-passed through a spring-loaded check valve, loaded to, say, ½ pound, so that with a slight increase of resistance of the strainers, the oil will by-pass them and the service never be interrupted.

Concerning arrangements of the turbines themselves, a design of a single complete expansion unit has already been referred to. The general design of the cross-compound units are shown in Fig. 11 and Fig. 12. The receiver pressure between the high and low-pressure elements approximates 5 pounds gage at full power, rendering it adaptable for connection to the auxiliary exhaust steam and the feed heater as has previously been discussed. The detail construction of these turbines, method of inserting blades, &c., &c., will not be dealt with in detail here as already much has been written concerning these,* except those details which apply particularly to marine work.

The turbines illustrated in Figs. 11 and 12 are essentially for vessels always operating at their maximum speed and are subject to modifications on ships where cruising speeds are involved. One of these, known as the "divided-flow" type, is shown in Figs. 13 and 14 and are the turbines applied to a Swedish cruiser. The combination comprises an impulse element in the high-pressure turbine through which passes all the steam to the system. The nozzles are arranged in groups, each under valve control so that the turbine may receive substantially full pressure at the nozzles with widely differing volumes of steam flow. The general arrangement of this combination is shown in Fig. 15. After passing the impulse element the steam divides, one portion continuing through the same turbine to condenser pressure; and a slightly greater portion passing over to the low-pressure element through which it expands to the condenser. For cruising

*See "Electric Journal" for January, February, March, April, 1918.

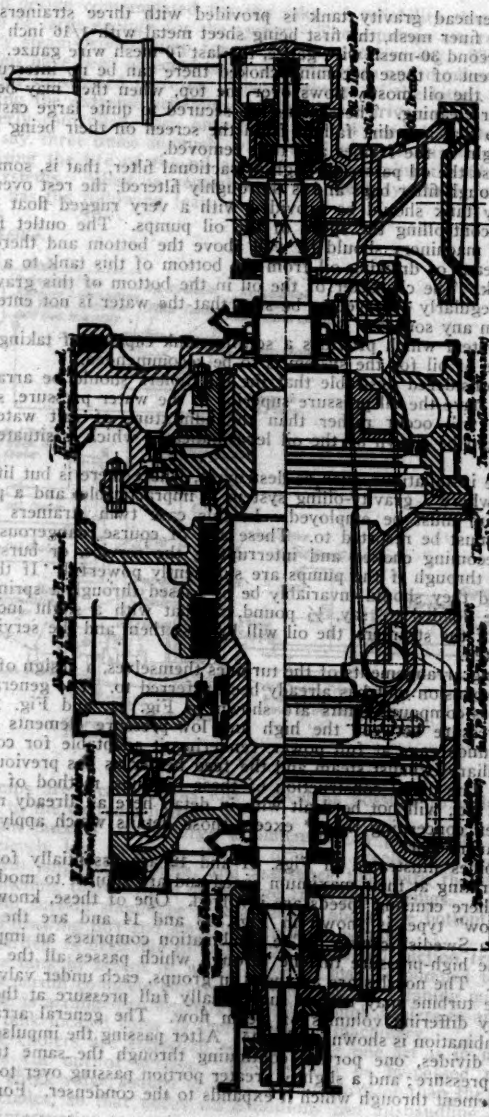


FIG. 11.

*See "Electric Journal" for January, February, March, April, 1913.

Fig. 72. L.P. ELEMENT OF CROSS-COMPOUND TURBINE

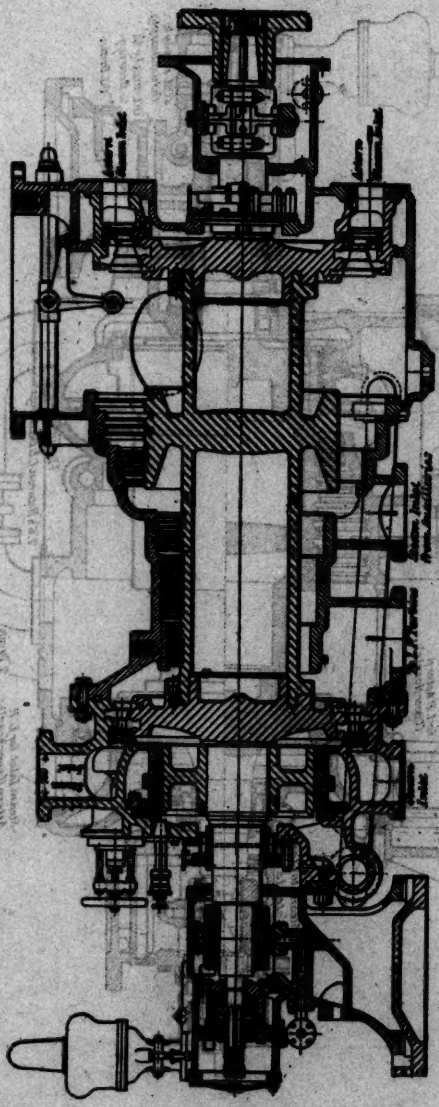


FIG. 13.

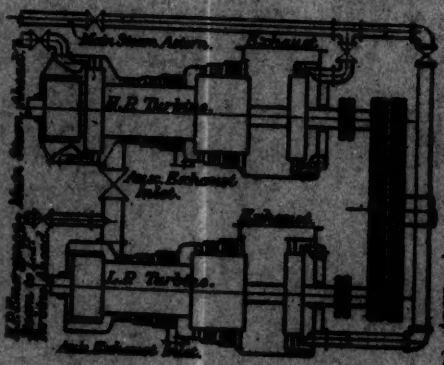
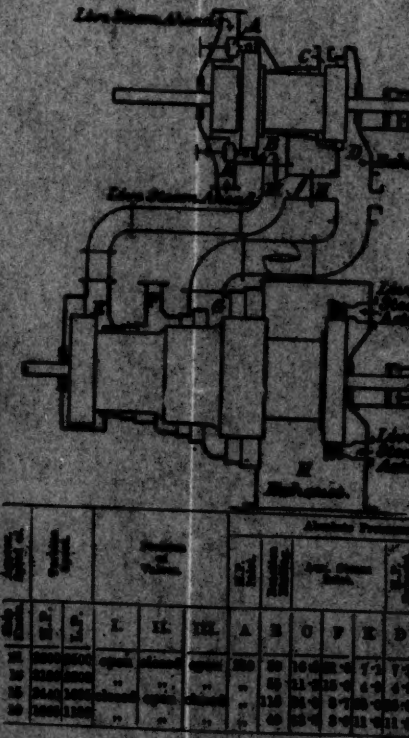
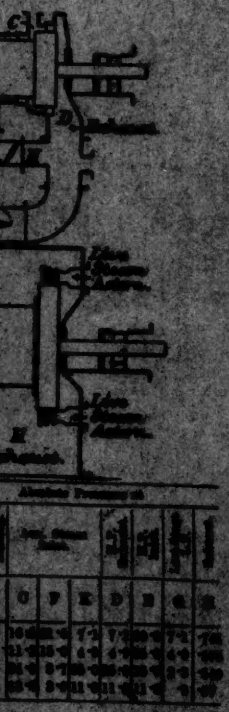



FIG. 15.




| | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | | | | |
| | | | I. | II. | III. | A | B | C | D | E | F | |
| 10 | | | | | | | | | | | | |
| 15 | | | | | | | | | | | | |
| 20 | | | | | | | | | | | | |

FIG. 16.



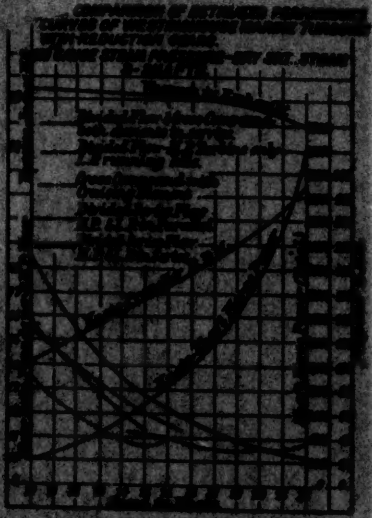


Fig. 17.

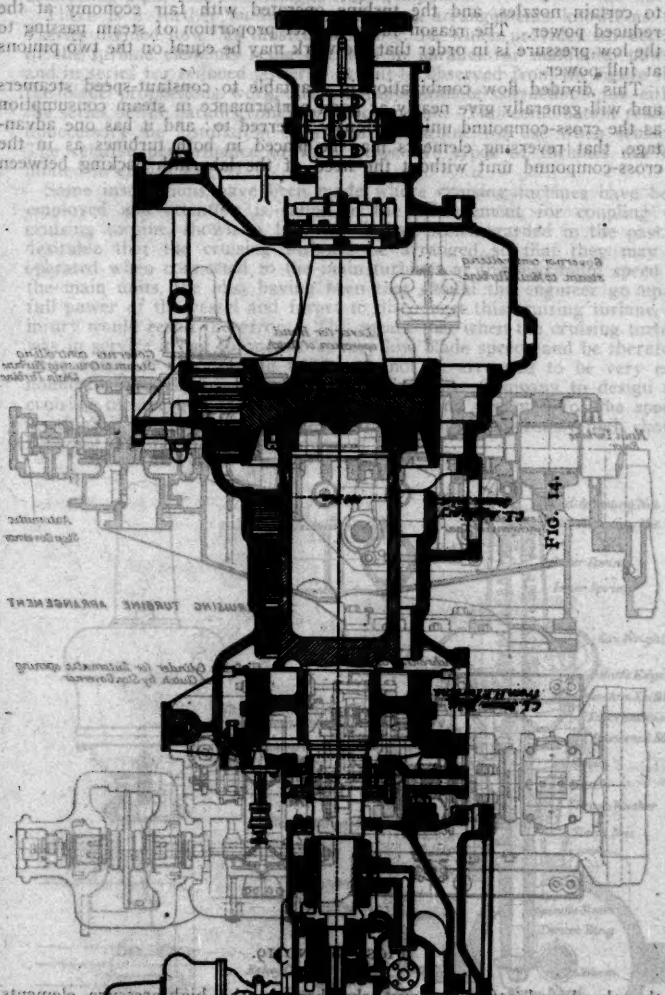
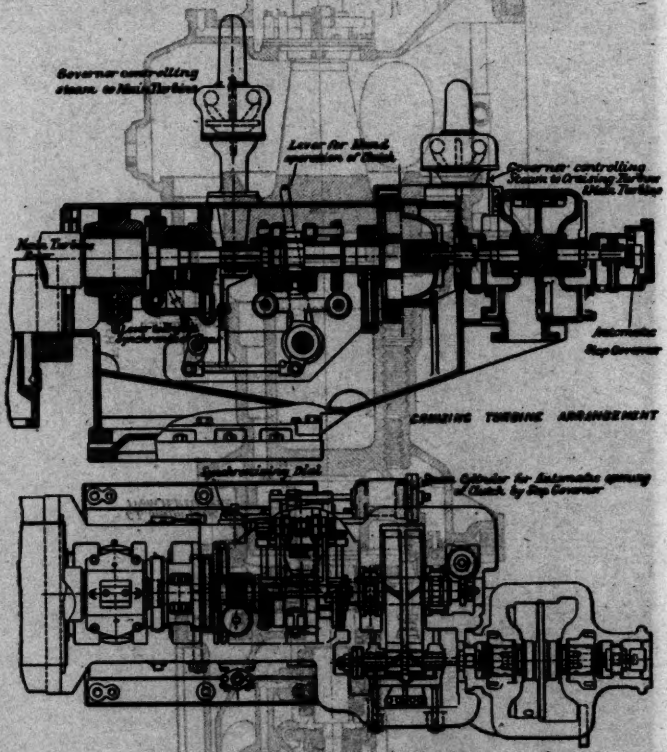


FIG. 14.

the one low-pressure element may be cut out of service, the steam closed to certain nozzles, and the turbine operated with fair economy at the reduced power. The reason for a greater proportion of steam passing to the low pressure is in order that the work may be equal on the two pinions at full power.

This divided flow combination is adaptable to constant-speed steamers and will generally give nearly as good performance in steam consumption as the cross-compound unit previously referred to; and it has one advantage, that reversing elements may be placed in both turbines as in the cross-compound unit without the need of the labyrinth packing between




FIGS. 18 AND 19.

the ahead and astern exhaust chambers in the high-pressure elements. Although not being compounded one with the other, the reversing elements are not as economical as the cross-compound unit first described. At full power the steam passes in parallel through three sets of low-pressure blades (if the low-pressure turbine be double flow), thus permitting smaller turbine diameters or higher rotative speeds, or both if desired, without reducing the adaptability of the turbine to expand to low

pressures. This type of installation has the advantage of smaller cross-connecting pipes than the straight cross-compound.

There is a further modification of these arrangements known as the "series divided flow" as illustrated in Fig. 16, which provides for certain of the turbine elements being operated in parallel for maximum powers and in series for reduced powers. It will be observed from the figure that the blade proportions work out with approximate corrections for either the series or the parallel condition of operation. This combination renders unnecessary the installation of a separate turbine for use when cruising. The relative performance of these respective types of turbines are exhibited in curves Fig. 17.

Some installations have been made where cruising turbines have been employed and attention is called to the arrangement for coupling the cruising turbine, shown in Fig. 18. It has been regarded in the past as desirable that the cruising turbines be arranged so that they may be operated when connected to the main turbines at the maximum speed of the main units, the idea having been that should the engineer go up to full power of the vessel and forget to disconnect this cruising turbine, no injury would result therefrom. This meant that when the cruising turbine was in service it had to operate at very low blade speeds and be therefore quite heavy or the cruising turbine cannot be arranged to be very economical. It is the practice of the Westinghouse Company to design the cruising turbine geared to the main turbine and designed for the speeds at which it is required in service which would be dangerously high should it be not disconnected at full power.

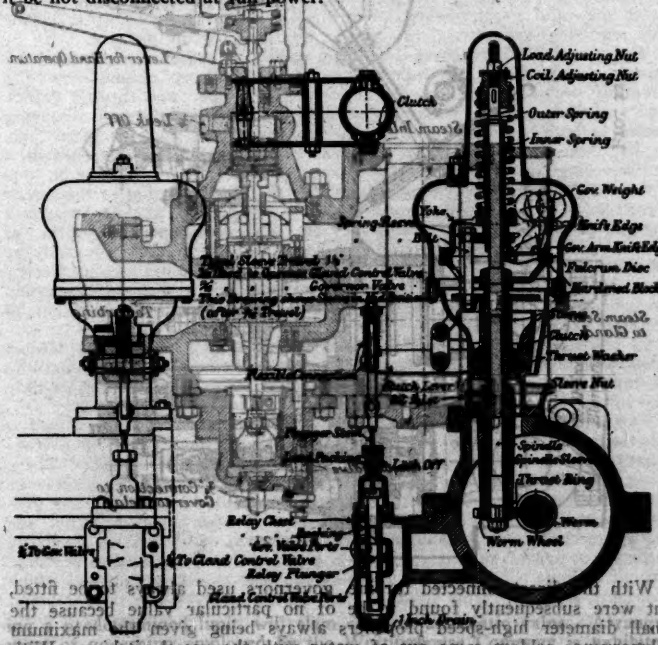



FIG. 20.

The design shown in Figs. 18 and 19 provides that should the cruising turbine operate above a predetermined speed, steam will automatically be shut off from it and the clutch automatically opened. The arrangement of this installation provides that the steam maneuvering valve is employed whether the cruising turbine is in operation or not. Means are provided for readily synchronizing the cruising turbine to the speed of the main turbine and throwing the clutch without affecting the speed of the vessel, provided it is running at speeds corresponding to that at which the cruising turbine should be operated.

Concerning steam turbine details that differ from land practice because of their application to ships, there are the following:

*Governing Arrangements.*—With the old reciprocating engines, racing in heavy weather was always to be reckoned with. While governing arrangements were sometimes furnished by the builder, they usually passed most of their existence in a store-room locker. While the reciprocator may accelerate rapidly, it may generally run at proportionately higher relative speeds than the turbine without injury.

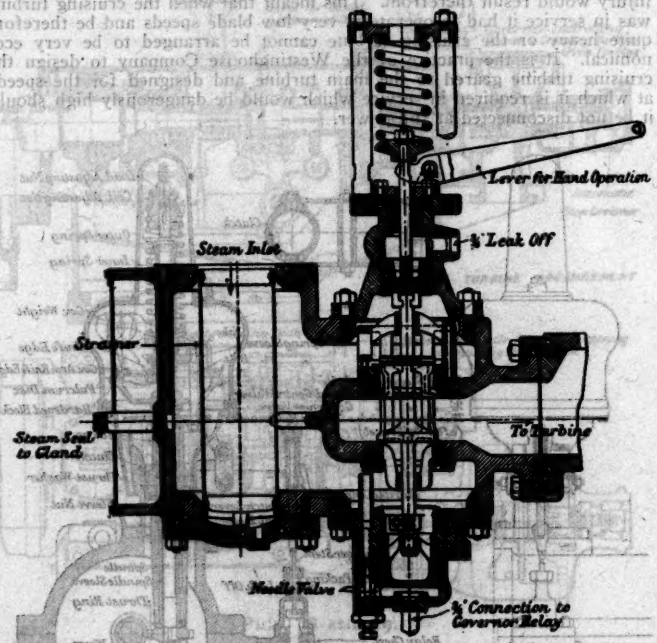



FIG. 21.

With the direct-connected turbine, governors used always to be fitted, but were subsequently found to be of no particular value because the small diameter high-speed propellers always being given the maximum submergence seldom came out of water with the vessel pitching. With the geared turbine the propeller dimensions revert to that of the recipro-

cating engine, and while, because of its inertia the turbine and gear may accelerate less rapidly than the reciprocator, very material increased speeds would be destructive. Therefore, a very dependable regulating mechanism is desirable which must at least have some semblance of being able to regulate, and must not be a mere stop governor which, on the speed reach-

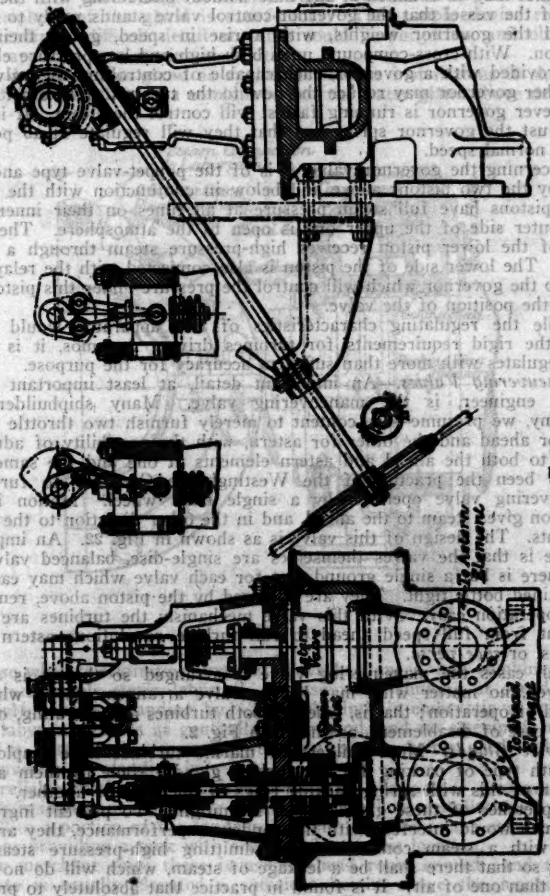


FIG. 21.

ing a predetermined limit, will slam shut an automatic valve. Arrangement of governing employed by the Westinghouse Company is exhibited in Figs. 20 and 21.

The governor is of the ordinary fly-ball type, the weights being carried on knife edges, and all working parts arranged to be under forced lubrication. The governor controls the valve shown in Fig. 21 by means of the steam relay, so that the connection between the turbine and the governor valves is merely a pipe, thus eliminating the necessity of connecting links, levers, etc. The relay may be moved by hand at any time by depressing the small spring that is above the relay. In this manner assurance may be obtained at any time without interfering with the operation of the vessel that the governor-control valve stands ready to control should the governor weights, with a rise in speed, go to their outer position. With cross-compound units both high and low-pressure elements are provided with a governor, each capable of controlling the valve, that is, either governor may reduce the flow to the turbine, or in other words, whichever governor is running fastest will control the steam. It is usual to adjust the governor springs so that they will regulate at 10 per cent above normal speed.

Concerning the governor valve, it is of the poppet-valve type and actuated by the two pistons above and below in conjunction with the spring. Both pistons have full steam pressure at all times on their inner sides. The outer side of the upper one is open to the atmosphere. The lower side of the lower piston receives high-pressure steam through a needle valve. The lower side of the piston is also connected with the relay adjacent to the governor which will control the pressure under this piston, and hence the position of the valve.

While the regulating characteristics of this apparatus would hardly meet the rigid requirements for turbines driving dynamos, it is simple and regulates with more than sufficient accuracy for the purpose.

*Maneuvering Valves.*—An important detail, at least important to the watch engineer, is the maneuvering valve. Many shipbuilders—for economy, we presume—are content to merely furnish two throttle valves, one for ahead and the other for astern, with the possibility of admitting steam to both the ahead and astern elements at one and the same time. It has been the practice of the Westinghouse Company to furnish a maneuvering valve operated by a single hand wheel. Motion in one direction gives steam to the ahead, and in the other direction to the astern elements. The design of this valve is as shown in Fig. 22. An important feature is that the valves themselves are single-disc, balanced valves, so that there is but a single ground seat for each valve which may easily be maintained bottle tight. They are balanced by the piston above, rendering their operation easy. With this valve mechanism the turbines are easily brought from full speed ahead to full turbine revolutions astern in 20 seconds, or *vice versa*.

In all cases the maneuvering valve is arranged so that it is always employed, no matter what may be the valve arrangements or what the method of operation; that is, whether both turbines are operating, or only one in case of disablement, as shown in Fig. 2.

*Turbine Glands.*—Most builders of marine turbines have employed a labyrinth type of packing for the turbine glands, some of them a combination of this with spring packing rings of one kind or another. There is no pretence of these glands being steamtight. To prevent ingress of air, which would interfere with the condenser performance, they are provided with a steam connection for admitting high-pressure steam for sealing so that there shall be a leakage of steam, which will do no harm, rather than one of air. It is found in practice that absolutely to preclude any air leakage there must be a superabundance of steam admitted which will blow, in considerable volumes, into the engine-room. A turbine gland used to be regarded as one of the difficult problems of design detail until, in 1903, the Westinghouse Company devised what is known as the water

gland. It merely comprises a centrifugal pump runner operating in a casing which, if furnished with water at the axis, would raise that water to some 10 pounds higher pressure than the maximum pressure against which the gland is designed to pack. In service its periphery is furnished with water at some 5 pounds greater pressure than the maximum pressure at which the gland is to pack, thus providing a water annulus at the outer part of the gland, forming a hermetic seal which is very effective. For condensing turbines a water pressure of 5 pounds gage at the periphery of the gland is satisfactory. This simple device rendered an exceedingly difficult detail one that could be forgotten, and obviously has its application to marine installations.

The design of this gland as applied to marine installations is shown in Fig. 23, in which there is a combination of the water gland above described

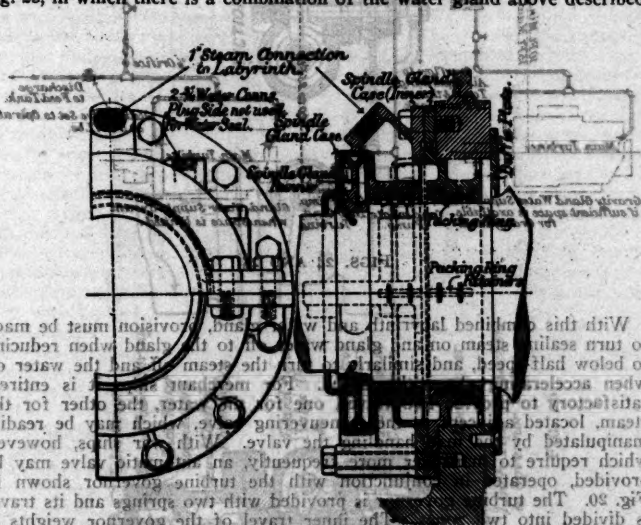

1" Steam Connection to Labyrinth
2-¾" Water Conns. Plug Side not used for Water Seal.
Spindle Gland Case
Spindle Gland Case (Inner)
Packing Ring
Packing Ring Gland

FIG. 23.

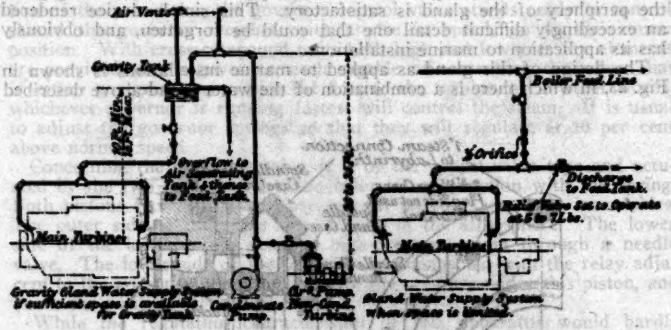
for use when operating at anything above half-speed, and combined therewith is a simple steam labyrinth for use when running at speeds below one-half or when standing by. This labyrinth is embryotic as compared with labyrinth glands as generally understood, and no attempt is made to have it particularly steamtight because of its being used only when maneuvering or standing by.

A reliable method of furnishing water at the proper pressure to the glands is important and easily arranged for in land installations by pumping all or a portion of the condensate to an overflow at the required elevation, or providing a tank with a float valve at this location.

For merchant ships there is usually head room enough, and the condensate is usually elevated a sufficient height above the machinery to furnish the required pressure, as shown in Fig. 24, and the system is quite automatic, requiring no adjustment.

On vessels where no head room is available it is customary, as shown in Fig. 25, to provide for taking a small quantity of water from the feed line in which an orifice is provided, which will pass but little more than

that required for the sealing of the glands. This water is carried to the gland system and there is provided a relief valve, by-passing the water not required back to the feed tank, thus maintaining the proper water pressure on the gland system. The gland casings may be removed for cleaning or inspection without otherwise dismantling the turbine.


Air Vents
Gravity Tank
Boiler Feed Line
Overflow to Air Condensing Tanks & Drains to Feed Tank
Discharge to Feed Tank
Main Turbines
Main Turbines
Gravity Gland Water Supply System. Overhead space is available for Gravity Tank.
Air & Pumps Non-Cond. Turbines
Gland Water Supply System where space is limited.

FIGS. 24 AND 25.

With this combined labyrinth and water gland, provision must be made to turn sealing steam on and gland water off to the gland when reducing to below half-speed, and similarly to turn the steam off and the water on when accelerating above this speed. For merchant ships it is entirely satisfactory to provide two valves, one for the water, the other for the steam, located adjacent to the maneuvering valve, which may be readily manipulated by the man handling the valve. With war ships, however, which require to maneuver more frequently, an automatic valve may be provided, operated in conjunction with the turbine governor shown in Fig. 20. The turbine governor is provided with two springs and its travel is divided into two zones. The inner travel of the governor weights is opposed by the light spring only, the governor weights being able to compress this spring at the speed at which the gland should change over from steam sealed to water sealed. The heavy spring comes into engagement at this speed, but the governor weights are unable to compress the two springs together until the turbine reaches the speed corresponding to which the governor should regulate the steam supply to the turbine.

Motion of the governor through the first part of its travel will, by means of the relay which it operates, either relieve or impose pressure in the chamber "X" of the gland control valve, Fig. 26, causing motion of the valve to either apply water or steam, or *vice versa*, according to whether the turbine is accelerating or retarding.

*Couplings.*—An important detail is the turbine coupling. To permit free motion of the floating pinion frame there is a flexible shaft provided between the turbine and pinion which provides the necessary element of flexibility. The turbine and gear must be capable of axial motion relative to one another. A coupling of the type shown in Fig. 27 is employed. Means are provided for disassembling either turbine or gear without reference to the other, and proper provision is made for lubricating the driving pins.

*Electric Drive.*—There is much difference of opinion concerning the relative merits of gear and electric motor drive. Discussion of this important matter is refrained from here in view of certain battleship installations now being carried out. Unquestionably these installations will be entirely successful, and will fulfil all expectations, but whether the added complication, increased cost and weight of machinery, together with increased steam consumption per propeller horsepower, are warranted, can be shown only by the development of the future.

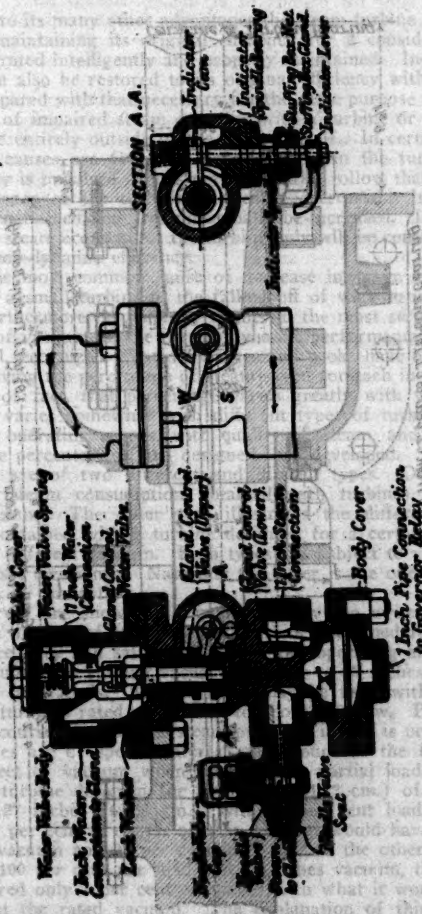



FIG. 26.

One system of electric motor drive suitable for the merchant marine has been proposed which has promise of economy greater than that hitherto attained. The proposition is to provide a number of relatively small Diesel engines whose cylinders shall not exceed a diameter of,

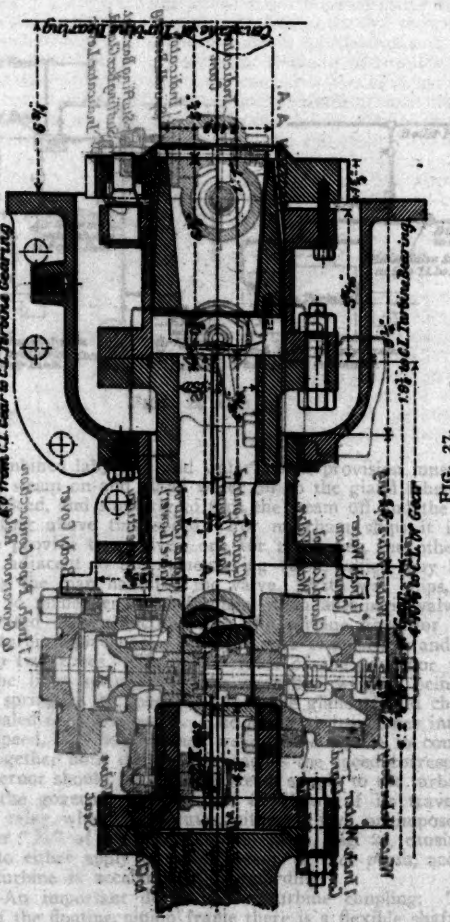


FIG. 27.

say, 10 inches, or within the limits where absolute reliability may be expected. These would be coupled to dynamos which in turn would operate a motor driving the propeller. The locations of the engines may be where desired without reference to the propelling motor.

## CAUSES OF IMPAIRED TURBINE ECONOMY.

By J. Y. Dahlstrand.

In addition to its many other advantages the steam turbine has the characteristic of maintaining its original efficiency for a considerable length of time if operated intelligently and properly maintained. In the majority of cases it can also be restored to its original efficiency with a small expenditure compared with that necessary for the same purpose on an engine.

The causes of impaired steam economy with a turbine or turbo-unit in many cases lie entirely outside of the turbine itself. In certain instances, however, the causes are found to develop right in the turbine. When steam economy is impaired it does not necessarily follow that the thermo-dynamic efficiency of the turbine affected is decreased. In some cases the thermo-dynamic efficiency might actually be increased. In the following article the steam economy of the turbine only will be considered, rather than the thermo-dynamic efficiency.

Probably the most common cause of decrease in steam economy, particularly with a small turbine, is the falling off of vacuum in the exhaust chamber. Furthermore, it generally results in the most serious loss.

The effect of vacuum on the thermo-dynamic performance of the steam turbine is well recognized. Various papers and books have been published with charts giving the percentage of steam saved for each inch of vacuum. As a matter of fact, that percentage varies greatly with varying steam pressure and varies somewhat with different types of turbines, with capacities, with operating speeds, with quality of steam, and, last but not least, with the percentage of the designed load developed.

The charts are of two different and distinct types. One shows the difference in steam consumption obtained with turbines designed for different vacuums. The other type illustrates the difference in steam performance obtained with a turbine designed for a certain vacuum but operated at a different vacuum. Both types are subject to variations, due to all the causes mentioned. Naturally the latter is the one which should be considered in this article.

Certain curves are shown in Fig. 1 made by the writer from actual test data on a number of 1000 kw. Kerr turbines designed for 150 pounds steam pressure (dry and saturated steam) and operating at 3600 revolutions per minute, with varying vacua (27 inches to 29 inches, or 68.58 cm. to 73.66 cm.) and loads. These curves may be used with fairly good accuracy for turbines rated at 500 kw. to about 3000 kw. Proper correction must of course be made for steam pressure, if this is not 150 pounds.

These curves are of special interest on account of the fact that they show the effect of vacuum when operating at partial loads. It will be noted that a turbine designed for 28 inches (71.12 cm.) of vacuum, and operating at 27 inches (68.58 cm.), with 25 per cent load, will have a steam rate 12 per cent in excess of that which it would have if operating at 28 inches vacuum and 25 per cent load. If, on the other hand, it was operating at 100 per cent per load and 27 inches vacuum, the steam rate will be impaired only 7 per cent compared with what it would have been if operated at the rated vacuum. The explanation of this lies, as will readily be understood, in the fact that when a turbine is operated at 25 per cent load the steam is throttled before entering the first-stage nozzles to a pressure very much lower than 150 pounds, and the number of heat units constituting the difference in available energy between 27 inches and 28 inches vacuum becomes a much higher percentage of the total energy available for use in the turbine.

Consulting the entropy heat diagram for a verification of the correction figure, 7 per cent, between 28 inches and 27 inches vacuum, it will be found that from 150 pounds to 28 inches vacuum there is 323 B.t.u. available, excluding the effect of reheating and considering straight adiabatic expansion. From 150 pounds to 27 inches vacuum there will be found 301 B.t.u. on the same basis. According to this there is 7½ per cent more heat available with 28 inches vacuum than with 27 inches vacuum. The correction figure for the latter water rate on this basis should be 7½ per cent.

There are certain factors which have a tendency to increase this correction figure. They are: First, the windage losses, which increase with the density of the steam; second, the fact that the nozzle passages in the last stages are too large for the lower vacuum, a circumstance equivalent to operating the last stages at a partial load. These circumstances, however, are generally more than outweighed by another fact.

As was mentioned before, an increase in steam consumption does not necessarily mean that the thermo-dynamic efficiency is decreased. On the contrary, it is sometimes actually increased. This happens to be true to some extent with nearly all commercial turbines of small and medium size. The thermo-dynamic efficiency is largely dependent on the "velocity ratio," or the relation between blade and steam velocities. For commercial reasons the turbine frames are actually in many cases somewhat smaller than those which would give the very best efficiency.

It might be stated here that future tendencies in steam-turbine building will probably follow somewhat different lines from recent practice. Instead of rating a turbine frame at its maximum capacity, without regard for a small loss in efficiency, indications are that the future policy will be to get the maximum efficiency even if at higher cost.

As has already been stated, low vacuum is far from uncommon, particularly in small power plants. It is not unusual to find that a turbine designed for 28 inches (71.12 cm.) of vacuum is actually operating at 25 inches (63.5 cm.) vacuum for months, owing to leaks in the exhaust line or some similar cause, with a loss of no less than 16 per cent in steam consumption. The same percentage loss will be found in coal consumption if the low vacuum is due to air leaks, as this does not lighten the work of the auxiliaries, but rather increases it.

The falling off of vacuum in the exhaust chamber of the steam turbine may be due to a number of different circumstances. The foremost of these are air leaks. These may develop in the exhaust piping between the turbine and condenser, in the condenser itself, or in the turbine glands. Air may also enter through the piston-rod and valve-stem packing on condensate pumps. In addition, air may enter the condenser through the circulating water.

The exhaust-end gland of the steam turbine is generally sealed with water or steam, the simplest form of this gland for condensing service being shown in Fig. 3. This gland is commonly used for impulse turbines. It has three carbon rings with a high-pressure connection. The valve throttling the high-pressure steam is opened sufficiently so that the pressure at the point A is high enough to keep the air from leaking in, and at the same time not high enough to cause any excessive leakage along the shaft towards the outside of the turbine. If too great an amount of high-pressure steam is necessary to seal the gland, it is evident that the packing needs to be replaced. Assurance that there is no leakage of air into the gland may be had by adjusting the seal valve so that a very slight mist of steam escapes along the shaft.

In certain cases with low-pressure turbines, where the steam supply comes from the exhaust of a non-condensing engine, steam leaks some-

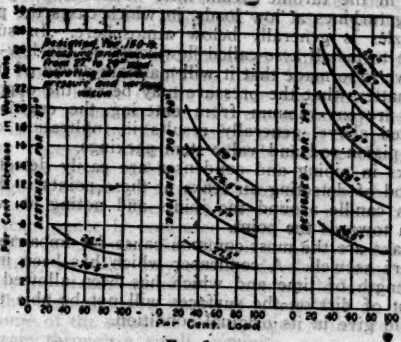
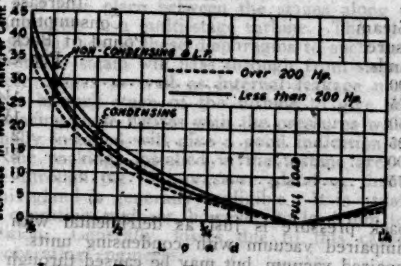



FIG. 1.




IMPAIRED TURBINE ECONOMY.—FIG. 2.

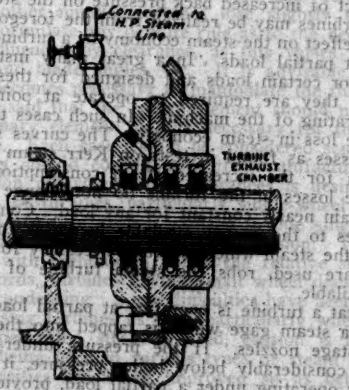



FIG. 3.

times develop in the turbine steam line. These are especially common when operating turbines at low loads, in which cases the pressure on the first-stage nozzles is generally below atmosphere. In such cases it is often advisable to install a so-called flow valve in the low-pressure steam line. This valve is so made that it will maintain ahead of itself a constant pressure, no matter what the pressure may be on the opposite side of it. It will hence prevent vacuum from entering the engine exhaust pipe, which, as stated above, generally results in infiltration of air through pipe joints or piston-rod and valve-stem packings.

The air leaks in piping and condenser are easily detected. The most common method of detecting air leaks at these points is that of bringing the flame of a candle around the joints and noting at what points the flame is drawn towards the point.

While air leaks are the most common cause for impairment in vacuum, there are others as well. A condenser which has been in service for a considerable length of time and which has been allowed to accumulate a large amount of dirt on the surfaces will not be able to maintain the vacuum it could give in its original condition.

| Initial Steam Pressure, Pounds. | Increase in Steam Consumption for Each Pound of Back Pressure. Per Cent. |
|---|---|
| 200 | 1 |
| 175 | 1¼ |
| 150 | 1½ |
| 125 | 2 |
| 100 | 2½ |
| 75 | 3 |

Excessive back pressure is just as detrimental with non-condensing machines as impaired vacuum with condensing units. This is not so common as impaired vacuum, but may be caused through carelessness on the part of operating engineers in leaving exhaust valves partly closed, etc. The effect of increased back pressure on the steam economy of non-condensing turbines may be realized from the foregoing table.

Next in its effect on the steam economy of a turbine comes the operation of turbines at partial loads. In a great many instances steam turbines are ordered for certain loads and designed for these loads. Later on it develops that they are required to operate at points very much below the full-load rating of the machine. In such cases there generally results a considerable loss in steam economy. The curves shown in Fig. 2 illustrate these losses as found in tests on Kerr steam turbines

The reason for the increased steam consumption at partial loads is obvious. The losses in the turbine do not change in proportion to the load, but remain nearly constant for any load. Consequently the relation of these losses to the energy available increases for partial loads. The throttling of the steam which is found necessary for partial loads, unless hand-valves are used, robs the steam turbine of some of the energy otherwise available.

The fact that a turbine is operating at partial load may be seen by observation of a steam gage which is tapped into the ring chamber ahead of the first-stage nozzles. If the pressure under normal operation at this point is considerably below boiler pressure, it is an indication that the turbine is operating under a partial load, provided, of course, that it was designed for only a reasonable pressure drop through the valve. The power developed with any type of turbine is very nearly proportional to the pressure at that point.

Turbines should be made to operate normally under the conditions for which they are designed. If it is necessary that they should carry overloads, overload valves—either automatic or hand-operated—should be used.

Up to this point only those circumstances which arise through steam conditions and from outside sources have been considered. Certain mechanical conditions inside of the turbine may also cause a thermodynamic loss in the steam turbine. Foremost among them is internal leakage. This is caused by the wearing out of the packing rings or bushings or the excessive clearance of labyrinth packing between the various stages in the steam turbine.

Internal leakage is, of course, most liable to develop where the pressure differences are very great. Therefore, in certain makes of turbines the high-pressure steam space is separated from the vacuum chamber with an internal gland. In cases of this kind it is always advisable to watch this gland carefully, as leaks may readily develop through it, causing a considerable loss in steam economy. Internal leakage is particularly detrimental in the high-pressure stages of a machine on account of the low specific volume of the steam in these stages, which results in a large discharge of steam through a small opening.

Leakage may also take place between the stages along the horizontal joints of the diaphragms in a multi-stage turbine. The steam and moisture may erode the surface of these diaphragms to such an extent that a considerable amount of steam will pass through from stage to stage.

Leakage between stages as well as internal leakage may be detected by placing a gage in each stage of the turbine to give the pressure in that stage. Comparing these figures with the pressures which the turbine had when originally installed will give a good indication as to the condition in which the packings installed in the turbine are.

Steam leakage through the high-pressure gland or through the leakage piping on this gland is generally negligible. It is usually sufficiently annoying to the operating engineer, however, to lead him to try to overcome this condition. Generally, losses due to this cause are greatly exaggerated by operating engineers.

Among the causes for impaired steam economy is excessive clearance between stationary and rotary elements. When this condition exists, Rateau, Curtis, and other impulse turbines lose slightly in power. It would be difficult to formulate any definite rule as to the degree in which steam economy is influenced by clearance, as this varies with capacity of machine, nozzle, and blade angles, etc. It is advisable, however, to bear it in mind if it becomes necessary to manipulate the thrust bearings. In general, it may be said that the rotary element should not be farther away from the stationary element than is necessary for the mechanical safety of the machine. Each case should be considered separately, however, so turbine users should consult with the manufacturers of their machines if information regarding axial clearances is desired.

With certain makes of turbines, particularly single-wheel impulse turbines, where the total pressure drop is utilized in one single expansion, considerable trouble has been experienced with erosion. This has been the case particularly with turbines in which the relative steam velocity was high—in other words, turbines having small wheel diameters or low speeds and operating condensing. Some difficulties have been experienced even with non-condensing turbines as well, particularly where the steam has been wet. If superheated steam is used, erosion is less common. Blade materials now being used in various types of steam turbines utilizing high steam velocities are being improved so as better to resist the erosive action of the steam. Erosion is absolutely unknown with other types of turbines on account of the low steam velocities employed. Through erosion of the blade edges the axial clearance between the

rotary and stationary elements is increased, which tends to cut down the power developed. Through the fact that these edges are made dull less perfect steam action is attained, which in turn causes a loss in power and efficiency.

The deposits of foreign substances in the turbine blades, blocking or partly blocking the passages, might be mentioned as another cause of impaired steam economy in turbines.

This trouble usually occurs when a turbine is being fed by one or more boilers which are giving about their maximum capacity. It is more serious with some types of boilers than others. In both fire-tube and water-tube boilers there is such active circulation of water that the mud and foreign material are prevented from settling in the boiler and are mechanically lifted to the top and carried through the steam lines.

In some steel-mill districts the deposit consists principally of a muddy material which is bound together by chemicals existing in the water. At the velocity at which steam enters the blades any slight material carried in the steam will be impelled against the blades with such a velocity that it will form a very compact deposit. In a great many instances the boiler-feed water supply contains large portions of vegetable matter, as well as boiler scale-forming materials. Where this is the case the combination produces a tough, rubber-like deposit which rapidly fills the turbine blades and causes serious trouble.

Owing to the immense quantity of steam passing through large turbines a rapid increase of deposits may result even though the amount of material mechanically carried over by the boiler is very small per unit of power. It is, therefore, frequently astonishing to note the accumulated deposits when the steam is apparently quite pure and clean. Sometimes turbine blades become so clogged that sufficient power cannot be produced and the turbine has to be dismantled and cleaned.

Steam that is very wet, when made from water containing scale-forming material, such as magnesia and other salts, will invariably deposit a scale-like substance over the turbine blades. This is somewhat distinct from the deposits mentioned previously. There are other deposits which are more local in character, such as, for instance, deposits which occur in paper-mill districts, where a considerable amount of pulp is discharged in a finely divided state into rivers adjacent. The result is that this finds its way into some neighboring power plant where, owing to its fine and light nature, it is carried over by the steam and very rapidly blocks up the turbine blades. Several cases have come to the writer's attention where this occurred, the turbine having to be dismantled at regular intervals and the blades cleaned out three or four times a year.

There are various means of combating the clogging of blades from the above-mentioned causes which are effective in different degrees. One which is frequently used, particularly for non-condensing turbines, is the placing of a lubricator in the turbine steam line. The oil vapor in the steam lubricates the blade surfaces and prevents the substances from becoming attached. Even for condensing turbines this method is frequently used, although it would appear not to be entirely advisable in installations where surface condensers are used, on account of oil deposits which would be found on the tubes. The writer knows of two instances where this method is used with turbines rated at 10,000 kw.

Undoubtedly the most effective means of preventing this trouble is that of installing a large receiver in the steam line. The steam velocity will be decreased greatly while passing through this receiver, and the foreign substances will be deposited on the walls. In addition to the receiver there should be installed near the turbine throttle a steam separator which will serve the double purpose of removing from the steam any remaining foreign substances and extracting a great deal of

moisture from the steam. It might in some cases be advisable to combine these two apparatus and install a so-called receiver-separator, in which case the foreign materials will be deposited on the baffles.—"Electrical World."

## THE "DEX" OIL COOLER.

In view of the developments which are taking place and the continued increase in the size of prime movers, the problem of efficiently cooling the large quantities of oil required for the bearings of turbines, thrust bearings and gear boxes—like many of the details in a large power installation—plays an important part in the smooth running of the station, and has compelled designers and those responsible for the running and maintenance of such installations to investigate the subject in some detail.

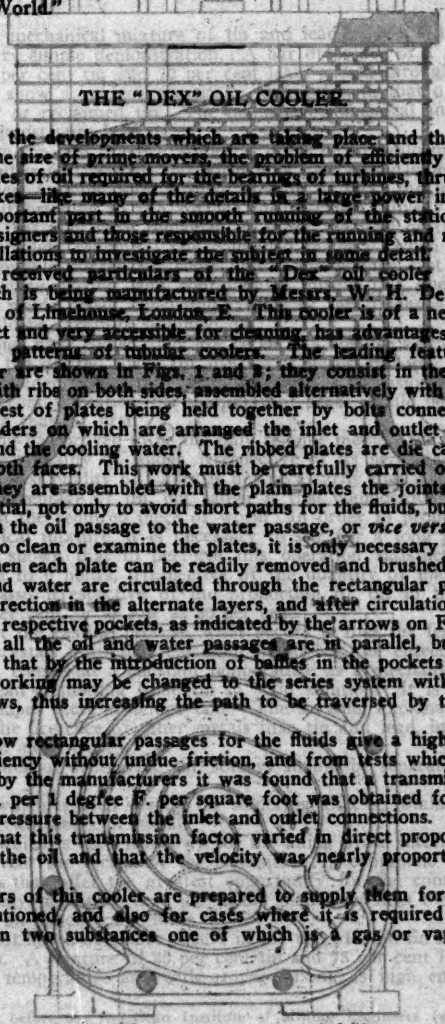
We have received particulars of the "Dex" oil cooler (Harrison's patent) which is being manufactured by Messrs. W. H. De Ritter and Co., Limited, of Limehouse, London, E. This cooler is of a new type and, being compact and very accessible for cleaning, has advantages over many of the older patterns of tubular coolers. The leading features of the "Dex" cooler are shown in Figs. 1 and 2; they consist in the use of flat plates cast with ribs on both sides, assembled alternatively with plain rolled plates, the nest of plates being held together by bolts connecting heavy cast-iron headers on which are arranged the inlet and outlet connections for the oil and the cooling water. The ribbed plates are die cast and then ground on both faces. This work must be carefully carried out to insure that when they are assembled with the plain plates the joints are sound. This is essential, not only to avoid short paths for the fluids, but to prevent leakage from the oil passage to the water passage, or *vice versa*. When it is desirable to clean or examine the plates, it is only necessary to withdraw the bolts, when each plate can be readily removed and brushed over.

The oil and water are circulated through the rectangular passages and in reverse direction in the alternate layers, and after circulation are delivered to their respective pockets, as indicated by the arrows on Figs. 1 and 2.

In Fig. 2, all the oil and water passages are in parallel, but it will be readily seen that by the introduction of baffles in the pockets the parallel system of working may be changed to the series system with two, three or more flows, thus increasing the path to be traversed by the oil to be cooled.

The shallow rectangular passages for the fluids give a high heat transmission efficiency without undue friction, and from tests which have been carried out by the manufacturers it was found that a transmission factor of 270 B.t.u. per 1 degree F. per square foot was obtained for a drop of 15 pounds pressure between the inlet and outlet connections. It appeared, moreover, that this transmission factor varied in direct proportion to the velocity of the oil and that the velocity was nearly proportional to the pressure.

The makers of this cooler are prepared to supply them for the services already mentioned, and also for cases where it is required to transmit heat between two substances one of which is a gas or vapor.—"Engineering."



SECTIONAL ELEVATION THRO A-B.

*Fig. 1.*

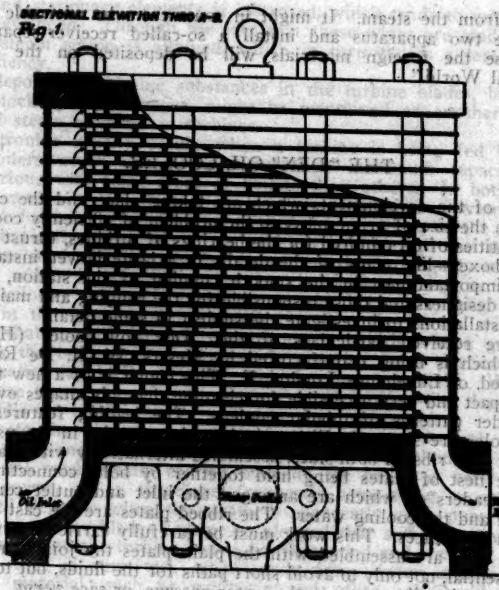
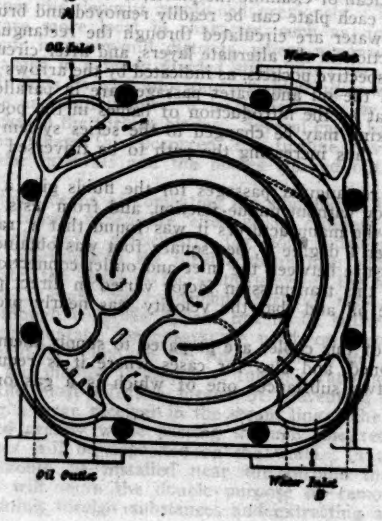


*Fig. 2.*



## SOLDER, ITS USE AND ABUSE.*

By Milton L. Lissberger,† New York, N. Y.

Solder is a mechanical mixture of tin and lead, a fact which is susceptible of very simple demonstration. A bar of solder of a grade even as low as 30 per cent tin and 70 per cent lead, passed through a buffing machine, will show a surface practically identical with that of a bar of second-quality or reclaimed tin. The buffings, on chemical analysis, will prove to be almost pure lead.

According to the best practice, solder is made in the following manner. Virgin pig lead is first melted, and when it is thoroughly liquefied, virgin pig tin is added, together with a small amount of flux; the latter is for the purpose of bringing to the surface the so-called "liver," consisting of impurities that may have remained in either the lead or the tin as a result of incomplete refining. The combined material, when completely liquid is thoroughly stirred for some hours, and is then cast into small pigs. Just before casting, and continuously during this operation, the molten metal yields dross, consisting largely of the oxides of lead and tin; this should be carefully skimmed off.

After the pigs have cooled, they are taken to a smaller kettle, remelted, and cast into the desired shape for use; or if wires, ribbons, etc., are to be made, the solder is cast into slugs suitable for extrusion and rolling. During this second operation, the skimming of dross should be even more carefully done than at first.

Hand mixing has proved to be the only reliable method for the production of the best quality of solder, irrespective of its percentages of lead and tin. The best quality of solder is not necessarily that which contains the highest percentage of tin, but rather is that composition which performs best on the required piece of work. In order to produce a thorough mechanical mixture, it is necessary to stir for a long period; experience has shown that to perform this operation satisfactorily takes from 5 hours to 6 hours, irrespective of the quantity of material being mixed, and also irrespective of the proportion of tin in the mixture, whether 60 per cent, or as low as 30 per cent.

Throughout the casting process, what occurs is that the lead solidifies in skeleton crystals until the remaining liquid has the eutectic composition, when it freezes at a constant temperature as a mechanical mixture of tin and lead containing some tin in solid solution. It is remarkable how many shapes these skeletons take. The seeming explanation of this variation is the presence of other metals than tin and lead, in very small proportion, or even traces.

With the idea of conserving tin, solders should be separated into two classes:

(1) That which is used strictly for soldering, that is, joining and holding together two pieces of metal.

(2) That which is used primarily for the filling of an interlocked joint, so as to prevent the escape of the contents of a container. It is these filling metals that offer the greatest opportunity for the conservation of tin. It is only necessary that the metal shall flow into the seam, and solidify into an impenetrable mass.

The greatest abuse of solder occurs in the use of high-tin mixtures for filling metals. A mixture of 25 per cent tin and 75 per cent lead, worked at the right temperature and with proper fluxing, is high enough in tin

*Paper read before the American Institute of Mining Engineers (slightly abbreviated).

†President, Marks Lissberger and Son, Inc.

for any filling purpose, as has been demonstrated in the practice of the oil canners, notably the Standard Oil Company.

The filling operation is usually conducted by machinery, but the users have frequently not realized that the baths are considerably richer in tin at the top, through which layer the container is being dragged, than the solder that is put into the baths. When the 40 : 60 solder, most commonly used on automatic can-making machines has not worked entirely satisfactorily, it has often been found that the addition of 1 inch or 2 inches to the depth of the bath has made the solder work very much better. Hence, one of the best means of conserving tin in can-making solder is to deepen all baths, whether on line machinery or for hand dipping, thus permitting the use of a lower-grade solder.

The fact that solder dross contains a higher percentage of tin than the original solder has usually been explained on the assumption that tin oxidizes more rapidly than lead. The probable explanation is, that in solder baths the lead is gradually working toward the bottom and the tin to the top, where it is exposed to the oxygen of the air, thus the oxide of solder is richer in tin than the original solder.

The overheating of solder is not only detrimental to the work, but also causes some, though not a very great, waste of tin through the excessive production of oxide. While this oxidation may be a source of considerable expense to the package manufacturer, it is not actually a very serious loss of tin because the reclaiming of these drosses, or oxides, has been so perfected that very little of the original metallic contents is lost. In these days, however, when every ounce of tin should be conserved, both to insure a sufficient supply for the most essential work, and to save the useless transportation of a material which comes such long distances by boat, overheating should be avoided, and all baths should be covered with a protecting material such as sal ammoniac, oil, charcoal, or ash.

In this connection, it should be emphasized that every particle of solder oxide should be preserved, and sent to the reclaimer. A teaspoonful of solder dross contains enough solder to make a 5-gallon can or 100 No. 1 cans. In many plants, even those of some of our largest consumers of solder, this dross is not collected and saved with sufficient care. It is seldom that a thorough cleaning and gathering together of the oxides takes place more than once a week.

Fire will purify these reclaimed materials, when properly refined, and in purity they will compare favorably with the virgin materials. However, too little attention has been paid to the proper refining of these scrap metals. Usually they have simply been put into a kettle, melted down, and then brought up or down to the required composition. This is not sufficient. Reclaimed metals are never equal to virgin metals, no matter how much refining they undergo; nevertheless, for certain classes of work they are economical and efficient. The repeated use of metal affects its physical permanency; yet the margin of safety in the use of solder is so large, and the length of time that solder is required to remain on the container is so comparatively short, that any lack of permanency can usually be safely disregarded.

After many years of experience, we have developed the following method of manufacture. The lead is first melted at a temperature which does not cause too rapid fusion. After the dross has had a chance to rise to the surface, it is carefully skimmed off before the required amount of tin is added, and slowly reduced to the liquid state. From the moment the tin is added, the solder is stirred by hand for 3 hours to 4 hours. A scavenger is then added and thoroughly worked for another 3 hours to 4 hours; the resulting dross is again skimmed, and the solder is cast into pigs of approximately 80 pounds each. These pigs are then re-melted in smaller kettles at a temperature which just causes free fluidity, and

the solder is then cast into the desired shapes. During the entire operation of final casting the caster stirs every time he takes a ladleful from the pot.

This work could be done much more rapidly at higher temperatures, and much more economically by the use of mechanical mixers, but the resulting solder would not be so thoroughly mixed, nor would it be so fluid. Mechanical mixing has a tendency to drive the oxide and dross back into the metals, thus diminishing the holding power of the solder. The scavenger must be chosen with great care, and the amount must be very accurately gaged; otherwise the scavenger becomes a constituent of the finished product, and, instead of being beneficial, is a detriment. We have found that in the grades of solder containing 46 per cent and less of tin, the addition of ½ per cent to ¾ per cent of the best grades of antimony increases the fluidity and holding strength of the solder for working tin plate.

Next to its use for containers, the largest consumption of solder has been on gasolene-motor radiators. The hand work on these radiators requires merely a free-flowing clean solder, but on the dipping work, where most of the solder is used, the greatest abuse has been practiced. As these radiators are composed of copper, low brass, or ordinary brass, no antimony whatever should be added to the solder used for this purpose. Also the affinity of tin and lead for zinc and copper will draw both of these metals from the radiators into the baths, and as both copper and zinc make solder sluggish, it does not take long (unless proper methods are employed for cleansing the baths) for the solder to become deteriorated.

These baths can be thoroughly cleaned by a mixture of rosin and sulphur, but as this operation produces very disagreeable black smoke throughout the plant, some method should be devised for disposing of it. When sulphur is used for removing zinc and copper, a sufficiently high temperature should be employed to insure the complete combustion of the sulphur. The baths should then be allowed to settle for at least half an hour after such heating, and the top carefully skimmed to remove any sulphides present. It is important to note that the presence of any non-metallic substance is injurious to solder, whether it has been added as a scavenger or is liberated from the original metals.

The question is frequently asked, what is the strongest solder that can be made. Numerous experiments have been made, but the results are confusing. Tests of tensile strength, based upon wires and cast bars, indicate that the higher the tin, up to 75 per cent tin, 25 per cent lead, the greater the breaking strength; in the case of two pieces of tin plate soldered together, the maximum strength is given by a solder containing around 42 per cent tin. Other tests were made on square 5-gallon cans, completely filled with water and then capped; when dropped from a height of about 100 feet the cans soldered with 46 per cent tin, 54 per cent lead, in no case broke at the seams, although the tin plate was ruptured. This was the only mixture that gave this result. Cans soldered with 47 per cent or more of tin, 53 per cent or less of lead, and with 45 per cent or less of tin, 55 per cent or more of lead, occasionally ruptured at the seams. These experiments were made most carefully and were afterward confirmed by subjecting the cans to air pressure.

I am thus inclined to believe that, in round figures, 46 per cent tin, 54 per cent lead, is the strongest mixture that can be used for general soldering purposes, particularly if ¼ per cent to ½ per cent of antimony be added to the mixture. The Bureau of Standards, with the approval of the War Industries Board, suggests that the highest grade of solder permitted should be 45 per cent tin, 55 per cent lead. For mechanical soldering, 40 per cent should be the highest tin ratio, and for most bath work it has been demonstrated that tin from 35 per cent to 38 per cent, according to the nature of the work, will give ample satisfaction, provided the solder is made properly.—"Engineering."

## ELECTRIC HEATED INDUSTRIAL FURNACES

By George J. Kirkgasser

War conditions have greatly increased the use of the electric furnace, not only in work where it was already used, but where it had hardly been given more than a little consideration. Because of the comparatively short time that the electric furnace has been commercially used, methods of operation have not been so well standardized, but with the number of men now familiar with them, there will be an interchange of ideas that will help toward standardization.

Where temperatures high enough to reduce steel to a molten condition are required the electric-arc furnace is employed, while for subsequent heat treatments, resistor-type furnaces are used, for in the latter process great accuracy and exactness of temperature control must be maintained.

The electric furnace is a metallurgical appliance in which any desired temperature up to the point of fusion of the best refractory materials obtainable can be attained and perfectly controlled. At these high temperatures chemical reactions take place much more rapidly than in other processes, and the most refractory metals and alloys can be reduced to fluidity.

An electric-arc furnace for steel making usually consists of a steel tank lined inside with refractory materials and fitted with working doors, spouts and tilting arrangements for teeming; regulatable carbon or graphite electrodes of a suitable section are inserted through the roof or walls. A high-tension electrical supply is transformed down to a low pressure for operating the furnace.

Arc furnaces are of two principal kinds; one known as the arc-radiation type, in which heat is radiated from the arc; and the arc-resistance or arc-conduction type, in which there is radiation of heat from the arc, and heat also generated as a result of the resistance due to the flow of the current in the furnace.

### Production of Electric Steel.

What are said to be the first patents on the electric arc steel furnace were recorded about 40 years ago, an Englishman, William Siemens, being the inventor. As with other patents, the patentee could not by any stretch of imagination realize to what extent the electric furnace would be used in the steel industry. The world's output of electric steel has multiplied more than tenfold since 1914. In that year there were about 30 electric furnaces in the United States and Canada, while at the beginning of 1918 there were 300 such furnaces in use in these countries with an annual output of about 2,000,000 tons.

The Illinois Steel Co. of South Chicago has the largest output of electric steel in the country, employing three 25-ton and two 15-ton furnaces of the Heroult type. The first, a 15-ton furnace, was installed in 1909. The products of this first furnace, steel rails, were laid on 14 railroad lines and the results showed that the electric steel was more ductile at low temperatures than either the openhearth or the Bessemer steel. Subsequent tests showed this also. Now the capacity of the plant is 16,000 to 17,000 tons of electric steel per month.

The electric furnaces manufacturing steel in England are now producing about 40 times the pre-war output, and it is now possible to do without the large imports of Swedish iron and steel which were formerly considered necessary to maintain the output of high-class products. Furthermore, the approaching exhaustion of the high-grade ores of

Cumberland may be looked upon with equanimity, as electricity makes possible the use of the inferior ores of Cleveland and the Midlands to manufacture steel of great purity.

The electric furnace has not only proved that it offers many advantages in steel making over the older processes, but has also shown that in making many classes of steel it is an absolute necessity.

Electric steel is free from dissolved gases which is due in a measure to the perfect control over the temperature of the steel by the operator. By raising the temperature until the steel is in an exceedingly fluid state, the gases are permitted to escape, after which the temperature can be regulated for casting so that the desired grain can be produced in the setting of the ingot.

In practically all cases the electric furnace is used not for the manufacture of steel directly from cold metal, but more as an electric refining process. Making steel directly from the cold metal requires a large consumption of current which makes the cost prohibitive. Usually it is used to finish metal previously made by the Bessemer or openhearth process.

Very often in steel plants the openhearth furnace is used for preliminary melting and an electric furnace for deoxidizing and desulphurizing, and for making any other changes needed to secure a steel according to analysis. In what is known as the "Triplex" system, the Bessemer converter is used for decarbonizing and desiliconizing the metal; the openhearth furnace is then used for dephosphorizing it and the electric furnace completes the work of deoxidizing and desulphurizing.

### BASIC OR ACID OPERATION.

The basic process of furnace operation is the more costly because of the present high cost of refractories. The use of basic slag serves to remove impurities from the metal, just as in the basic openhearth steel furnace. Phosphorus, oxygen and sulphur may be removed, which permits using relatively cheap raw materials. With acid operation no phosphorus or sulphur is removed and more care is needed to hold the elements between desired limits.

In lining basic furnaces, magnesite or dolomite is used up to about the slag line and above this silica, magnesia, and chrome brick is provided; while for the roof, silica is used.

For acid operation, silica is used for the entire lining. Acid operation has been used to a great extent because of the abundance of shell turnings, steel clippings, and punchings from steel shipyards, etc. It is a little quicker than the basic, as it is not necessary nor possible to refine the charge.

In Canada electric pig iron is made from steel scrap, the scrap being used in this way rather than being turned into refined steel, because of the shortage of low phosphorus pig iron. About 500 kw. hours of power per ton is average current consumption in this process.

Dr. John A. Matthews, president of the Halcomb Steel Co., Syracuse, N. Y., recently said:

"The electric furnace requires for successful operation less labor per ton than is necessary for crucible steel, but it calls for metallurgical skill in order to produce the best results. Easy oxidizing metals like vanadium, chromium and manganese, are readily handled and less of them required to give the content desired in the steel. Sulphur and phosphorus can be readily eliminated and the yield of sound steel increased.

"With the electric furnace, alloy steels are made in the furnace itself rather than in the ladle, and in this way there is better opportunity for increasing the solution, diffusion and homogeneity in the product. All

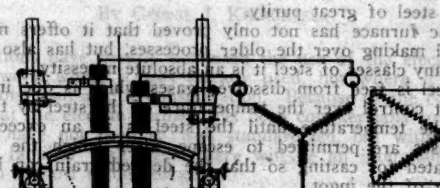
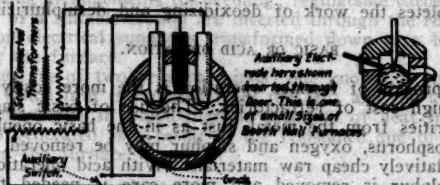
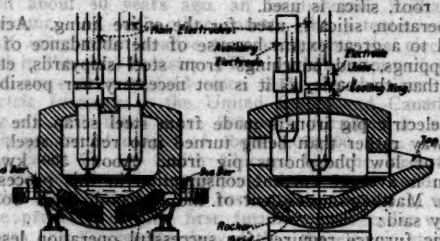


DIAGRAM SHOWING ELECTRICAL CONNECTIONS AND CURRENTS SET UP IN THE MOLTEN METAL IN A GREAVES-ETCHELL FURNACE.




ELECTRICAL CONNECTIONS OF BOOTH-HALL FURNACE WITH TWO-PHASE CURRENT SUPPLY.

AUXILIARY ELECTRODE INSERTED THROUGH DOOR



BOOTH-HALL FURNACE—TWO-PHASE CURRENT SUPPLY

of these things make for high quality, and quality is the first consideration. In addition to this, the electric furnace performs an economic function because of its adaptability for handling and recovering alloy values contained in scrap. This was of especial importance during the war, when alloys had to be conserved. The alloy content of chromium, manganese and vanadium in scrap used in the open-hearth steel is only recovered to a very small extent, and not only is the alloy value lost but the oxides formed are frequently a source of trouble in the final product."

The superiority of electric steel castings over those made by any other process has been conclusively proved by the extremely severe conditions to which they have been subjected under specialized war conditions. The castings produced can be finished off dead mild and a one-inch square bar can be bent as cast, no expensive annealing operations being necessary. The carbon contents usually vary between 0.2 per cent and 0.25 per cent, while the sulphur and phosphorus are each reduced below 0.015 per cent and the manganese and silicon are adjusted to suit the work.

The tensile properties of ordinary electric cast steel are:

Maximum tensile strength, 35 tons per square inch.
Yield point, 25 tons per square inch.
Reduction of area, 50 per cent.
Elongation, 30 per cent.

The maximum tensile stress can be readily increased to even 100 tons per square inch by the introduction of special elements such as nickel, chromium and vanadium.

Steel from the electric furnace is finished off under a reducing atmosphere of carbonic oxide and a slag out of which the metallic oxides have been reduced, and is singularly free from blow-holes when properly cast. This is due to the elimination of the gases that molten steel usually holds in solution and the ease with which the steel can be poured into small and intricate castings, flowing smoothly and setting perfectly.

Electric steel castings can be made as cheaply as malleable iron castings, while the former are lighter for the same strength.

### HEROULT FURNACES.

Heroult furnaces are of the arc-resistance type, the electrodes for introducing the current being placed above the bath and the current passing from one electrode to the metal and from the metal to the next electrode. The bath is thus heated by radiation from the arc and by the resistance of the steel and slag to the flow of current. The steel shell is lined with a refractory material, the nature of which depends on whether the furnace is to be basic or acid in its operation.

The 25-ton furnace of this type at the Duquesne plant of the Carnegie Steel Co. requires 175 kw. hours per ton of steel and operates basic on hot metal. The three electrodes are of 12-inch graphite and are used up to the extent of about four pounds per ton of steel. Three phase current received at 6,600 volts is transformed to 100 volts at the furnace. Six heats are made per day of 24 hours, and each heat averages about 2½ hours.

### GREAVES-ETCHELLS FURNACE.

In principle, the "Greaves-Etchells" Furnace is a combined arc-resistance furnace.

Two phases of a three-phase low-tension supply are connected to their respective upper graphite or carbon electrodes, while the third phase is connected to the bottom of the hearth. The current flowing through the hearth generates a considerable amount of heat immediately below the

liquid in the most efficient manner possible, while the electric arcs arranged over the bath maintain the slag and surface at the desired temperature.

The effect of this bottom heating is to cause convection currents in the molten metal, which insure a constant circulation and a uniform product. The outside of the furnace bottom remains cold, little or no heat being lost in this direction.

Charges of high-speed steel and castings have been made from cold scrap in one hour and ten minutes. Present sizes include 1, 2, 3, 5, 6, 10, 12½, 20 and 30-ton furnaces. A half-ton furnace is also made specially for high-speed steel, small steel castings and special tool steel.

### BOOTH-HALL FURNACES.

The Booth-Hall furnace is built for use on single, two and three-phase electric power lines, in the following capacities: ¾, 1½, 3, 6, and 10-ton. Power is introduced to the furnace through one, two, or three main vertical electrodes, depending on whether the furnace is single, two, or three-phase. An auxiliary electrode is also introduced and there are cast-steel grids imbedded in the hearth with a refractory facing. The Booth-Hall furnace is therefore of the vertical arc type with conducting hearth.

In starting a heat with a cold charge in the two-phase Booth-Hall electric furnace, a pawl on the auxiliary electrode holder is released, permitting the auxiliary electrode to rest upon the scrap charge. Arcs are then drawn with the two main electrodes, and the metal begins to melt under the arcs. The auxiliary electrode acts as a common neutral, or return, no arcs being drawn and, therefore, no metal melting under the auxiliary electrode. When a molten bath has accumulated upon the hearth, sufficient to make the hearth conductive of electricity, ammeters indicate that current is beginning to flow through the cast-steel grids. The auxiliary electrode is then raised up out of contact with the charge and plugs its own hole in the roof. The furnace then continues to operate with the current entering the furnace through the main vertical electrodes and being carried through the bath and the hearth to the grids, and so out through the grids. The connections from the transformer to the furnace are so arranged that each of the two phases has an independent circuit.

At the West Michigan Steel Foundry, the three-ton furnace has averaged 2 hours 15 minutes per heat and the current required per net ton is about 676. Records of the analysis for the months of August and September, 1918, are given. The operations are all on a basic furnace lining.

The Snyder furnace is also of the arc-resistance type several of which are in operation in the plant of the Haynes Stellite Co. of Kokomo, Ind.

The Rennerfelt furnace is of the arc-radiation type, used mostly in 1-ton and 3-ton sizes in various industries as foundries, machine makers, valve makers and some in the non-ferrous industry.

### AVERAGE ANALYSIS.

| Element. | Per Cent. |
|---|---|
| Carbon | 0.25 |
| Manganese | 0.55 |
| Sulphur | 0.046 |
| Phosphorus | 0.045 |

AVERAGE PHYSICAL PROPERTIES.

| | |
|---|---|
| Tensile Strength | 63,040 pounds. |
| Elastic Limit | 38,010 pounds. |
| Elongation | 29.9 per cent. |
| Reduction in Area | 41.5 per cent. |

ELECTRIC FURNACES IN FOUNDRIES.

Electric furnaces are today in operation in foundries for melting steel scrap, refining steel scrap, refining molten steel and melting gray iron, malleable iron, ferro-manganese, ferro-alloys, copper and copper alloys. In some foundries the electric furnace is used for the melting and refining of steel, where an extra high quality of steel is desired, low in sulphur and phosphorus and meeting definite specifications. The Canadian and United States governments have both installed a considerable number of electric furnaces for this purpose, the chief object being the production of steel low in sulphur. Eventually the application of the electric furnace to the melting of malleable iron will be as important as it is to the steel industry today. Ordinarily, the electric furnace cannot compete commercially with the cupola for melting gray iron. It is useful, however, in manufacturing aeroplane motor cylinders of very high-grade gray iron, owing to the facts that the sulphur content is kept down, that the metal is melted under conditions ideal for the production of iron free from air-holes, and that the iron produced from an electric furnace has the same density that is characteristic of electric steel.

Mr. W. E. Moore, president of the Moore Electric Furnace Co., said at the recent Metal Congress and American Foundrymen's Convention in Milwaukee, that basic-lined furnaces were most suitable for the ordinary steel foundry although at present more acid-lined furnaces are in use because specifications are liberal as to sulphur and phosphorus content. High-grade scrap has been available because of the manufacture of munitions. The acid-lined furnace is considered simpler and cheaper.

Because of the ease of operation of the electric furnace, small foundries can now make high-grade steel castings, replacing iron castings to a great extent. In addition there is a wide field opening up for alloy steel castings of a grade that can be properly made from electric-furnace steel. Such castings can be heat-treated and may in some cases replace drop forgings.

HEAT TREATMENT OF METALS.

Heat treatment, electrically came into its own with a vengeance during the past two years. Crankshafts for Liberty motors, bolts and nuts for airplanes, gears for our army trucks, cast-steel anchor chains for the emergency fleet, gun parts, locomotive axles have all been treated in electric annealing furnaces. One furnace is 193 feet long, used for annealing bright strip steel, the guarantee being that the steel will come out of the furnace as bright as it went in. The strip is fed through on cars which move continuously through the furnace, the capacity for a day being 150 tons.

Heat treatment includes "hardening," "annealing," and "drawing" or "tempering." The furnace temperatures vary from 1,550 to 1,650 degrees F. The accurate control possible with the electric furnace makes it easy to provide exactly the temperature required, and this is important because if the temperature is too low, the results are not obtained; while if too high, the steel is damaged—"burned."

Electric means of measuring and the pyrometer make it possible to arrive at a standard, and to duplicate the conditions. This eliminates waste and improper results. The heat from the electric furnace is also uniform and the temperature of the metal is raised uniformly.

The resistance electric furnace has the usual firebrick walls and insulating material enclosed in steel frame work. The resistance elements, located entirely within the furnace chamber, are rectangular in cross section and run the entire length of the furnace, or, in the case of circular furnaces, entirely around the furnace. This trough is composed of a carbide of silicon material. The electrodes for leading the current through the furnace walls are of carbon or graphite. The resistance material is laid in the refractory trough mentioned and is composed of broken carbon or graphite. The current flows from electrode to electrode through this carbon resistor, which heats up in the manner of the ordinary carbon-filament lamp, the heat radiating from the carbon and the trough heating the furnace chamber, and, in turn, the material to be heated. The control of the heat input is obtained by means of various voltages from the secondary of a special transformer, a relatively definite number of kilowatts being obtained from each of the various voltages.

Hearth-type annealing furnaces are adapted for small pieces as gears, nuts, bolts, etc. The Baily furnace of this type has a capacity for heating 200 pounds of steel to a temperature of 1,620 degrees F. in one hour with a current consumption of 400 kw. hour per ton of metal heated. The temperature can be held within limits of five degrees variation.

With larger furnaces of the car and pusher types, greater economy of current is obtained.

Automatic operation has attracted the widest attention because of the results obtained. A heat-treating equipment of this kind was referred to in a paper read by Mr. Baily, president of the Electric Furnace Co. of Alliance, Ohio. This consists of two furnaces, each having a hearth 20 feet long and 7 feet wide. In the treatment of 14,000 tons of steel not a single piece of material was rejected due to faulty heat treatment.

These furnaces are automatically controlled by contact-making pyrometers, which, in conjunction with a special electrically-operated valve pulpit, control all the operations of the furnace. The pyrometer couples located in the discharge end of the furnace are dominated by the temperature of the material itself. Therefore when the material at the discharge end has reached the temperature required, the pyrometer couple actuates the pyrometer instrument, which, in turn, through suitable relays, sets in motion the valve pulpit device, which in proper sequence raises the door of the charging furnace, swings the manipulator platform under the door of the furnace, and by means of the pusher pushes in a fresh charge of material at the same time discharging on to the manipulator platform the material located under the pyrometer couple which starts the operation. When this material has become centered on the charging platform, it is removed by the manipulator from the furnace and plunged into the quenching bath as the door closes.

At this period of the operation a time-element device is thrown into circuit, which holds the material being quenched in the bath a definite time, after which the manipulator raises the material out of the quench and places it in front of the charging platform of the second furnace, where it remains at rest until the pyrometer mechanism of the second or "drawing" furnace puts into motion a similar operation to that just described, on which the heat-treated material is discharged from the second furnace.

Besides the use of electric furnaces in the making and treatment of steel, they are also employed for treating brass castings, silver, copper, phosphor and manganese bronze and other non-ferrous metals. Arc-type

and resistance-type reverberatory furnaces are used. There are a number of furnaces in the experimental stage for this class of work.

For the melting of alloys as nickel, steel and nichrome, the Driver-Harris Co. is successfully using a 2-ton Heroult arc furnace operating at about 2,200 degrees F.

A plant of five electric furnaces have just been put in operation by the Anaconda Copper Co. for the production of ferromanganese from low-grade manganese ore.

In the manufacture of Stellite, an alloy developed especially for cutting tools, the electric furnace seems to meet the requirements. A resistance type was tried and later three Snyder arc-type furnaces installed. Power consumption averages about one-third kilowatt per pound of metal melted, the melting time required being about 30 minutes.

HOSKINS FURNACES.

Small electric furnaces of the type generating heat by means of passing current through a resistor or heating element are used for many purposes, such as ash determinations, enameling (as watch dials in watch factories), heat treating (as of dies, cutting tools, etc.).

In the Hoskins type F C furnace the current enters by way of one of the electrodes, passes along one side through the carbon plates, through a connector plate, and back to the other electrode through the other carbon plates. The heat is radiated uniformly onto the work. The carbon-plate type furnaces are for work such as for high-speed tool steel where the temperatures required are above 1,900 to 2,000 degrees F.

For temperatures around 1,800 degrees F. the heating element is an alloy wire, chromel. Furnaces of this kind are used for the usual purposes mentioned above and for pre-heating high-speed steel before treating in the carbon plate type or type F C furnace.

By means of pyrometer and regulating rheostat the temperatures are controlled at all times, which makes electric furnaces ideal. A pyrometer with thermo-couple for inserting in the furnace and connecting cord are shown in one of the illustrations.

The regulation of the temperature requires merely a regulation of the current. This is a very simple matter and accounts for the ease and accuracy of control of the electric furnace. Where a rheostat is used, the turning of the rheostat handle provides the regulation. Particularly with larger furnaces, the regulating transformer is preferred, although it can only be used where alternating current is employed. Since low voltage currents are used in the operation of small furnaces, alternating current is preferable, since by means of a transformer the voltage of the usual 110, 220 or 440-volt system can be reduced to the lower voltage required which may be from 10 to 50 volts. Direct-current furnaces are, of course, made and can be used on the usual systems where the voltages are 110 or 220.

The Edison Appliance Company furnace is used as an annealing furnace for treatment of carbon steels, for blueing, annealing and preheating high speed steel and dies, and is used for porcelain enameling of small parts. The operating temperature is about 1,800 degrees F. When put in use about 3,000 watts are consumed, but this is maintained only for part of an hour, when the current consumed drops to less than 1,000 watts which is sufficient to keep the heated furnace up to its proper temperature.

For laboratory and research work the small-tube furnace which operates at a current consumption of 350 watts, and will maintain a temperature of 1,800 degrees F. for a continued period is used extensively. It may be

used for checking pyrometer couples, ash determinations, organic and inorganic combustion work.

Crucible-type electric furnaces are used in determining the decalescent and recalescent point of steel, melting alloys, ash determination, and general research laboratory work.—"Industrial Management."

## H. M. T. B. DESTROYER *MOUNSEY*.

Destroyers, as is well known, have played a leading part in the defeat of Germany at sea. Not merely did they get home again and again on the German Fleet as it fled from the Jutland battle, but the destroyers have been the terror of the U boats. For obvious reasons it has not been possible to publish until now any details of the more recent of our additions to these craft, but the declaration of the Armistice has relaxed the restrictions in vogue, and we are thus enabled to give an illustration and some particulars of the destroyer *Mounsey*, the boat which, under the command of Lieutenant Craven, saved, under circumstances of very great difficulty, no less than 696 lives when the *Otranto* was torpedoed on October 6 last. The sea at the time was exceedingly rough, and it would have been fatal to have brought a lightly-constructed vessel like the *Mounsey* alongside of the cruiser. The saving of life was therefore effected by maintaining the *Mounsey* under way, so that she passed the *Otranto* within a few feet, allowing the people to jump from one ship to the other. The maneuver had to be repeated many times, and was, under the circumstances, a very original and successful method of rescuing those endangered without great risk to the destroyer herself.

The *Mounsey* was built by Messrs. A. F. Yarrow and Co., Ltd., at their Scotstoun works, and on her measured mile trials attained a speed of over 39 knots. This trial was run with the boat fully armed and equipped, and with sufficient fuel on board for a run of 1,000 miles at an economical speed. Further particulars are given below.

| | |
|---|---|
| Length between perpendiculars | 260 feet 3 inches. |
| Length overall | 271 feet 6 inches. |
| Beam | 25 feet 7½ inches. |
| Depth, midships | 16 feet 3 inches. |
| Total heating surface | 22,017 square feet. |

Four hours' trial—

| | |
|---|---|
| Draught forward, 8 feet 1½ inches; draught aft, 8 feet 2¾ inches | = 835.3 tons at yard. |
| Speed, 4 hours | 38.605 knots. |
| Speed on measured mile | 39.018 knots. |
| Revolutions per minute, 4-hour trial | 685.6 |
| Revolutions per minute on measured-mile trial | 693.02. |
| Oil consumed in 4 hours | 57.33 tons. |
| Load on trial | 158 tons. |
| Oil fuel capacity | 238 tons. |
| Radius of action at full speed | 615 miles. |

Armament—

Three 4-inch quick fires; two 2-pounders; two twin 21-inch torpedo tubes.

Complement—79.

PARTICULARS OF RUNS ON MEASURED MILES.

| No. | Hour. p. m. | Boiler Steam. | Vacuum. | Air Pressure. | Revolutions. | Time on Miles. | Speed. |
|---|---|---|---|---|---|---|---|
| 1 ... ... | 1.45 | 259 | 28 | 6.5 | 689.0 | 1-32.4 | 38.962 |
| 2 ... ... | 1.54 | 257 | 28 | 6.5 | 692.0 | 1-32.4 | 38.962 |
| 3 ... ... | 2.4 | 257 | 28 | 6.5 | 693.0 | 1-32.2 | 39.056 |
| 4 ... ... | 2.13 | 258 | 28 | 6.5 | 693.5 | 1-32.0 | 39.130 |
| 5 ... ... | 2.24 | 258 | 28 | 6.5 | 694.0 | 1-33.0 | 38.710 |
| 6 ... ... | 2.34 | 259 | 28 | 6.5 | 692.75 | 1-31.4 | 39.388 |
| Mean on measured mile ... ... | | 258 | 28 | 6.5 | 693.023 | | 39.018 |
| Mean in 4 hours ... | | 258 | 28 | 6.4 | 685.6 | | 38.605 |

It may be added the first Italian vessel to enter the port of Trieste was the *Audacé*, which was built by Messrs. Yarrow and Co. eighteen months ago.—"Engineering."

## THE COMPOSITION AND PROPERTIES OF STEELS.

By Howard Ensaw.

In most branches of the engineering trade it is at times necessary to reduce the weight of some mechanism, and this applies particularly to the manufacture of automobiles and aeroplanes. This forms probably the chief reason for a thorough study of the nature and properties of the materials employed in the construction of such mechanisms.

It is essential that the designer should have a full knowledge of the strength and durability of all the materials to be used, and it is important that those materials should be of a specified composition, and that the treatment should be checked by suitable tests in order that the results in actual practice shall correspond with the data on which the designer worked.

Until about four years ago the number of firms with a knowledge of the composition of aircraft steels and experience in their heat treatment was most limited, and today there are so many brands of alloy steels requiring different treatment that great care must be exercised. As the hardening and tempering process is liable to error due to the number of brands available, users are well advised to choose only a few types of steel to cover their requirements. The treatment of the selected steels quickly becomes familiar, and the possibility of error is reduced.

A range of about ten steels will cover the requirements of practically every firm, and the object of these notes is to give the composition and properties of a range which might be chosen with advantage.

As variation in heat treatment is required with different steels in order to obtain the best results, it is obviously necessary to consider the mechanical properties along with the chemical composition.

It will be seen that it is possible, however, particularly in the case of nickel-chrome steels, to obtain widely varying results with varied heat treatment.

The table has been compiled from actual tests made with a particular make of nickel-chrome oil-hardening steel, and illustrates this point very well.

The material which gave the above tests when treated in the manner indicated forms one of the most interesting metallurgical studies, and is of exceptional value in aircraft work. It will be noticed that various degrees of hardness can be obtained, and this factor is made great use of in the manufacture of articles where distortion is a disadvantage and at the same time difficult to avoid.

The articles are first rough-machined and then heat-treated to a Brinell hardness of 250 to 321. At this hardness the articles can be machined, and we then have no distortion in the finished goods, as there is no subsequent heat treatment.

| O. H. Degrees C. | Temp. Degrees C. | Time. Minutes. | T. S. Tons per square inch. | R. A. per cent. | B. N. |
|---|---|---|---|---|---|
| 825 | 650 | 30 | 59 | 57 | 269 |
| 825 | 600 | 30 | 64 | 48 | 286 |
| 825 | 550 | 30 | 69 | 42 | 321 |
| 825 | 500 | 30 | 75 | 43 | 332 |
| 825 | 450 | 30 | 85 | 42 | 375 |
| 825 | 450 | 20 | 87 | 49 | 364 |
| 825 | 400 | 30 | 96 | 38 | 418 |
| 825 | 350 | 30 | 104 | 33 | 430 |
| 825 | 200 | 15 | 121 | 38 | 512 |
| 825 | Not | tempered | 123 | 34 | 477 |
| 825 | Not | tempered | 126 | 21 | 512 |
| 800 | Not | tempered | 127 | 27 | 477 |

O.H. = Temperature in degrees centigrade at which sample was hardened in oil.

Temp. = Temperature in degrees centigrade at which sample was tempered.
Time = Time in minutes for tempering.
T.S. = Tensile strength in tons per square inch given by test piece.
R.A. = Reduction of area registered during above test.
B.N. = Brinell hardness number.

It will thus be seen that it is possible to obtain widely varying results from one brand of steel, and it might also be said that it is possible to obtain similar results from two steels of widely different chemical composition.

It is highly advisable that the steel manufacturer should coöperate to a greater extent with the steel user to obtain a better understanding of the work for which the finished article is intended. Items subjected to different stresses require different materials and treatment, and in the case of resistance to wear it is well to bear in mind that a steel containing a low proportion of nickel and a low proportion of chromium will resist wear better than a steel containing a high proportion of nickel and no chromium.

It has become generally recognized that it is not only necessary to utilize materials of a sufficiently high yield point, but the materials must also be of a nature suitable for resisting wear if the articles are subjected to abrasion, and able to withstand shock if the articles are subjected to vibration. The advisability of putting samples of the material through tests which bear some resemblance to the stresses which will be met in

actual use is obvious, and in connection with aero-engine manufacture this question has been given considerable attention.

Perhaps the first problem to be attended to in the manufacture of parts which are highly stressed is the question of the composition of the materials employed. It will be understood that if the original material should be of incorrect proportions satisfactory results can scarcely be expected.

It is expensive to introduce the nickel, chromium, manganese and vanadium into mild steel, and unless the alloy steel thus produced is carefully watched its value may be destroyed as the result of incorrect proportions of carbon, sulphur and phosphorus. A full chemical analysis from time to time will probably save the reputation of the firm concerned.

In addition to the expense of manufacture, alloy steels require more careful treatment. A scientifically arranged system of checking temperatures is necessary if the results are to be relied upon. As the employment of these steels is necessary when we require an ultimate strength which may reach 130 tons per square inch, the expense of materials and testing apparatus must necessarily be met.

The elements previously referred to have varying effects, and are introduced one or more at a time. The alloys differ over a wide range, and the treatment of these various alloy steels must be modified to suit each particular composition.

When nickel is added to plain carbon steel, the skin of the hardened material is harder, and the depth to which the quenching effect penetrates is greater.

Chromium and tungsten added to plain carbon steel enable a more highly finished surface to be obtained when machining, and a finer grain is produced in the metal; but these elements do not appear materially to influence the hardness or the depth of case of the hardened material.

Chrome-vanadium steels do not appear to have given quite as successful results as nickel-chrome steels, but for certain purposes they are preferable. The presence of vanadium increases the depth of the hardening effect, but when vanadium is present in even medium quantities the material in the annealed state is very hard, and machining difficulties arise.

Nickel is perhaps the most valuable element to the metallurgist, and by simply normalizing a bar of nickel steel to the same tensile strength as a sample of medium carbon steel a comparison of the results readily establishes the superiority of the alloy steel. Increased elasticity is obtained, and the impact test figures are considerably improved, so that in cases where elasticity and shock-resisting qualities are required, nickel steel is preferable, providing the carbon content is not excessive when the proportion of nickel is considered. If the carbon content is not within the limits specified later, the probability is that the article would fracture under a lighter load than plain carbon steel would stand.

Bearing in mind the influence of these alloys when introduced into steel, we can proceed to inquire into the composition of the materials, and it must be noted that it is necessary to maintain the proportions of the elements within certain limits in relation to each other.

The use of plain carbon steel in aircraft work is most restricted, as we generally require material able to withstand higher stresses; but it can be used for certain parts which are not subjected to great shock or wear, such as tachometer gear. It is then possible to use a steel suitable for automatic machinery, and material to the following specification will be found to give satisfactory results:

| | |
|---|---|
| Carbon | Not greater than 0.25 per cent. |
| Manganese | Not greater than 0.85 per cent. |

| | |
|---|---|
| Silicon | Not greater than 0.20 per cent. |
| Phosphorus | Not greater than 0.06 per cent. |
| Sulphur | Not greater than 0.06 per cent. |

Should it be required to use a slightly higher tensile steel the following proportions will be found satisfactory, but machining is more difficult and the tools will require more frequent attention:

| | |
|---|---|
| Carbon | Not greater than 0.40 per cent. |
| Manganese | Not greater than 1.00 per cent. |
| Silicon | Not greater than 0.20 per cent. |
| Phosphorus | Not greater than 0.06 per cent. |
| Sulphur | Not greater than 0.06 per cent. |

None of the above steels is suitable for casehardening, but as they can be obtained in bright drawn bars the advantages of their use in all possible instances are obvious.

Although casehardening nickel steel has in a great many cases taken the place of casehardening mild steel, the latter is still used to a great extent in the manufacture of valve tappets, engine-timing gears, camshafts, gudgeon pins, etc. Parts made from this material present a good resistance to wear, and two qualities, low and medium carbon, may be used with advantage. The low carbon will give a minimum of 23 tons breaking strength when normalized at 900 degrees to 920 degrees C., and the composition as follows:

| | |
|---|---|
| Carbon | 0.10 per cent. |
| Silicon | 0.18 per cent. |
| Manganese | 0.60 per cent. |
| Sulphur | 0.04 per cent. |
| Phosphorus | 0.04 per cent. |

When this material is normalized as above the physical tests should be as follows:

| | |
|---|---|
| Breaking strength | 23 to 28 tons per square inch. |
| Yield ratio | Not less than 50 per cent. |
| Elongation | Not less than 30 per cent. |
| Reduction of area | Not less than 50 per cent. |
| Brinell hardness | 92 to 112. |

This material if well made will be found to caseharden well and uniformly, but it is frequently necessary to employ a similar steel with slightly improved breaking strength, although there is a tendency for the elongation to diminish.

This second-quality casehardening mild steel can be specified as follows:

| | |
|---|---|
| Carbon | 0.15 per cent. |
| Silicon | 0.18 per cent. |
| Manganese | 0.75 per cent. |
| Sulphur | 0.06 per cent. |
| Phosphorus | 0.06 per cent. |

It will be noticed that higher proportions of sulphur and manganese are permissible in this instance. When the above material is normalized at 890 degrees to 920 degrees C., the following tests should be obtained:

| | |
|---|---|
| Tensile breaking strength | 25 to 33 tons per square inch. |
| Yield ratio | Not less than 50 per cent. |
| Elongation | Not less than 25 per cent. |
| Reduction in area | Not less than 50 per cent. |
| Brinell hardness | 103 to 143. |

The normalizing temperatures given are within 50 degrees C. above the critical temperature in each case. The carburizing should take place at 900 degrees C., the length of time depending upon the depth of case required, and the articles should then be allowed to cool down in the compound. In order to refine the structure of the core the articles should then be heated to about 870 degrees C. and either allowed to cool in the air or quenched in water. Finally they should be reheated to 780 degrees C. and quenched in water to harden the case.

With certain makes of steel not conforming correctly to the above specifications it may be necessary to depart very slightly from the above temperatures, but the most suitable conditions can readily be ascertained by trial and a little advice from the steelmaker.

There are many instances where it is necessary to have some knowledge of the mechanical properties of the core of casehardened articles, and this can be obtained by allowing test pieces to go through the same heat treatment and at the same time as the parts being manufactured, taking care to turn off the case to the depth to which the carbon has penetrated. It will generally be found that by the refining influence of the heat treatment the yield point and the breaking strength are improved, and the elongation and reduction of area are reduced.

In carrying out some recent tests in connection with the last material detailed, the tests on the material as rolled and after the hardening treatment were as follows:

| | As Rolled. | After Treatment. |
|---|---|---|
| Breaking strength | 31 tons | 36 tons. |
| Yield point | 21 tons | 25 tons. |
| Elongation | 28 per cent | 23 per cent |
| Reduction of area | 58 per cent | 51 per cent. |

These figures are of some value, as it must be realized that we cannot count on the case to assist in withstanding any shocks, and our calculations must take into consideration only the extent to which the material may be loaded based on the strength of the core only.

Should circumstances arise where it is necessary to employ material of higher ultimate strength and better surface-wearing qualities than the casehardening carbon steels detailed above, use can be made of casehardening nickel steels, and here it is necessary to exercise greater care in manufacture and closer accuracy in heat treatment.

A good low-nickel alloy steel can be made up as follows:

| | |
|---|---|
| Carbon | 0.13 per cent. |
| Silicon | 0.25 per cent. |
| Manganese | 0.45 per cent. |
| Sulphur | 0.04 per cent. |
| Phosphorus | 0.04 per cent. |
| Nickel | 2.00 per cent. |

The carbon content must be carefully checked, as, if excessive, this causes shortness, and attempts to straighten any parts distorted in hardening will result in fracture.

This material when normalized at 850 degrees to 900 degrees C. should give the following results:

| | |
|---|---|
| Tensile breaking strength | 25 to 35 tons per square inch. |
| Yield ratio | Not less than 55 per cent. |
| Elongation | Not less than 30 per cent. |
| Reduction of area | Not less than 55 per cent. |
| Brinell hardness | 103 to 153. |

A step higher in tensile breaking strength and yield ratio can be obtained with an alloy steel containing a larger proportion of nickel, and the elongation will remain about the same. A chemical analysis of such a steel should show the results given below, and the necessity for maintaining a low proportion of the injurious elements renders the manufacture of this steel somewhat difficult and consequently expensive:

| | |
|---|---|
| Carbon | 0.15 per cent. |
| Silicon | 0.18 per cent. |
| Sulphur | 0.04 per cent. |
| Phosphorus | 0.04 per cent. |
| Manganese | 0.35 per cent. |
| Nickel | 5.00 per cent. |

The casehardening nickel steels given above are suitable for articles where a good hard wearing surface is necessary, but when quenching to harden the case it is necessary to reheat to a somewhat lower temperature than is the case with carbon steels, and when it is decided which brand of material to use it is advisable to carry out a series of tests to ascertain the most suitable temperature. Occasionally it is found necessary to manufacture articles where good wearing qualities are required, but the parts are so thin that a hard case would be disastrous, as there would be no core left. In such instances an alloy steel containing about 3 per cent of nickel, 0.30 per cent of carbon, and about 0.70 per cent of manganese will be found of great value, as it is thus possible to obtain 45 tons per square inch ultimate stress and a Brinell hardness of about 200.

The materials which have probably been of greatest value to the automobile manufacturer engaged on aero-engine work are probably the alloy steels containing nickel and chromium. These are very high in tensile strength, and are suitable for oil-quenching and tempering or air-hardening according to the proportions of nickel and chromium in their composition.

These alloys can be so constructed that they can first be hardened and then tempered to give almost any degree of hardness between 250 and 500 Brinell, and it is not a difficult matter to bring the hardness to about 320 Brinell. Up to this figure the material can be machined, and the parts will stand a great deal of shock and wear; but should the hardness be greater, the probability is that the further machining would be impossible.

It must not be concluded from the above remarks that the Brinell hardness mineral can be taken as a general guide to the machining properties of materials, but in comparing two samples of the same steel it can be taken as a rough indication. When comparing different alloys it is quite possible that one steel can easily be cut by another which registers lower on Brinell test.

A high tensile nickel-chrome steel suitable for tempering to various degrees of hardness can be built up as follows:

| | |
|---|---|
| Carbon | 0.20 to 0.30 per cent. |
| Silicon | Not over 0.30 per cent. |
| Manganese | 0.35 to 0.60 per cent. |
| Sulphur | Not over 0.04 per cent. |
| Phosphorus | Not over 0.04 per cent. |
| Nickel | 2.75 to 3.50 per cent. |
| Chromium | 0.45 to 0.75 per cent. |

If the above material be heated up to 820 degrees C. and quenched in whale oil, then tempered at 600 degrees C., the test results should be approximately:

| | |
|---|---|
| Tensile breaking strength | Not less than 45 tons per square inch. |
| Yield ratio | Not less than 5 per cent. |
| Elongation | Not less than 15 per cent. |
| Reduction of area | Not less than 50 per cent. |
| Brinell hardness number | 179. |

When using this material for a high tensile and a hardness of about 320 Brinell as previously suggested, it is an advantage to rough-out the parts before heat treating, so that very little metal is left for removing in the hard state.—"Mechanical World."

## NEW TYPE OF MARINE OIL ENGINE.

Mr. Carl W. Weiss, associated with the Metz & Weiss Engine Company for a number of years and now connected with the Weiss Engine Company, 17 Battery Place, New York, has developed a surface-ignition, medium-compression, two-cycle oil engine that promises to widen the scope of the heavy oil motor. The engine illustrated herewith is a 400-brake horsepower, 4-cylinder unit, although the type is also being built with six and eight cylinders. Among other features it possesses a new method of scavenging, and, in this connection, when it is recalled that the ordinary type of surface-ignition, medium-compression oil engine uses a baffle plate in conjunction with crank-case compression for scavenging—a construction which, due to uneven thicknesses of metal, makes for uneven contraction and expansion that frequently results in cracked piston heads—it is well to note that in the accompanying illustration the piston is conical in shape and a rather wide departure from usual practice.

### BONE OF CONTENTION.

The bone of contention regarding the relative efficiency of the surface-ignition, medium-compression, two-cycle oil engine has always been the matter of scavenging. That previous existing types of this style engine have not completely burned their charge or completely cleared the cylinder of burned gases after each explosion has been the contention. In the engine here shown there has been incorporated an entirely new method of scavenging. There are three annular sets of piston-controlled ports: (1) the exhaust, (2) the supplementary, and (3) the crank-case port. The supplementary ports are open to either atmospheric or under low pressure of air supplied by a small pressure blower. As the piston uncovers the first series—the exhaust ports—near the end of the expansion stroke, the pressure of the cylinder drops to atmosphere, and, due to the abrupt discharge and the forcible cooling of the gases, the pressure at once goes

down to several points below atmosphere. At this point the supplementary ports open, allowing a charge of pure air to sweep in radially over the conical piston head, displacing the exhaust gases left in the cylinder, while immediately following this, as the crank moves through the lower dead center, the crank-case air under approximately five pounds pressure per square inch also flows in over the conical piston head by way of the annular series of ports formed by the spirally ribbed lower parts of the cylinder liner. In this way three completely separate and distinct charges of air are introduced into the cylinder during the scavenging process, which, undoubtedly, accounts for the fuel efficiency of this new type and its ability to operate indefinitely without undue heating of piston head.

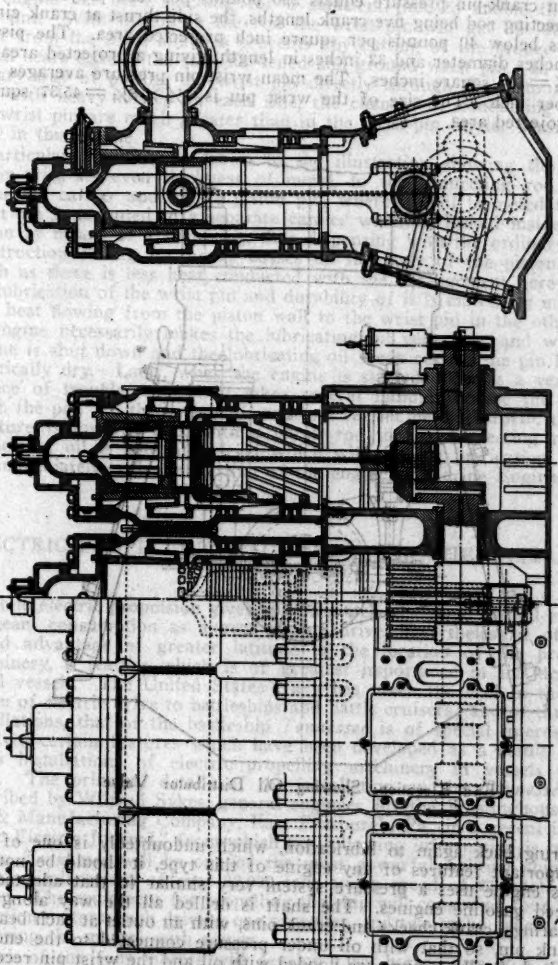
### SIMPLE OIL INJECTION SYSTEM.

The oil-injection system of this multi-cylinder engine is simple. In place of direct-driven governor-control injection pumps, there is an independent duplex pump to keep the oil under constant high pressure, and a compensating distributor valve, driven from the engine shaft, arranged for timing adjustment for different grades of oil and either direction of rotation. This pump is connected to the air receiver used for starting and reversing the engine. With a normal air pressure of 200 pounds in the receiver, the oil pressure is kept at 1,000 pounds by a reducing valve in the air line. Heavy oils require high pressures for efficient spraying. The governor is designed to act directly on the compensating valve and is, in fact, carried by the distributor valve and submerged.

There is a spiral gear mounted on the front end of the crank shaft which drives the oil distributor on one side and the air distributor for starting and reversing on the other. Each cylinder has an air check valve piped to the air distributor and a relief valve open to the atmosphere. These relief valves can be operated either independently or simultaneously by a lever at the front end of the engine, so that the entire control of speed, starting and reversing, and pressure relief is brought within easy access of the engineer.

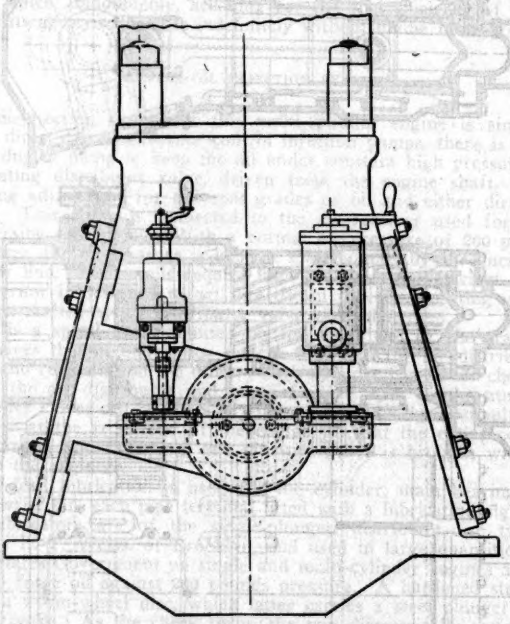
Forced-feed lubrication is used for the cylinder, main bearings, crank pin and wrist pin, each pipe terminal fitted with a lubricating sight check. These lubricators are of the single-plunger, distributor-disc type, furnished by L. T. Weiss, of Brooklyn, and used in large quantities by the United States Government on single and multi-cylinder engines and guaranteed to force oil against 200 pounds pressure. A hardened steel worm engages a worm-wheel disc, which latter carries a steel plunger with its operating yoke. As the check turns, the two diametrically opposite projections of the yoke and the fixed star wheel operate as an escapement, reciprocating the plunger, to draw in and discharge oil through a hole in the disc, registering alternately with the suction and discharge hole in the base. Each discharge has a screw coupling for copper tubing. The whole mechanism being very simple and substantial, and submerged in oil, with an extremely slow movement, these lubricators run for many years without the least wear or adjustment.

This type of marine engine requires no flywheel and no special scavenging pumps. The weight per horsepower is reduced to approximately 150 pounds without sacrificing a reliable factor of safety within the limits of working pressure. At a pressure of 500 pounds per square inch the maximum main bearing pressure does not exceed 509 pounds per square inch projected area. The shaft in the 16½ by 22 unit is 8¼ inches diameter. The bearing length is 12¾ inches. There is a center bearing between each cylinder in the multi-cylinder type. The bearing pressure, therefore, is $500 \times 214 = 107{,}000$, divided by $210 = 509$ pounds, as stated above, while



Elevation and Cross Section of White Engine, Showing Conical Pistons and Cylinder Liners, Together With Method of Supporting Main Bearings from Side

the mean pressure is below 100 pounds per square inch projected area. The mean crank-pin pressure equals 290 pounds per inch projected area. The connecting rod being five crank lengths, the side thrust at crank circle tangent is below 40 pounds per square inch projected area. The piston is 16½ inches diameter and 33 inches in length, giving a projected area of $33 \times 16\frac{1}{2} = 544$ square inches. The mean wrist-pin pressure averages 475 pounds per inch. The size of the wrist pin is $5\frac{1}{2} \times 8\frac{1}{4} = 45.37$ square inches projected area.



End Elevation, Showing Oil Distributor Valve

Referring back again to lubrication, which undoubtedly is one of the most important features of any engine of this type, it should be noticed that this engine uses a pressure system very similar to that adopted by high-speed gasoline engines. The shaft is drilled all the way along the main bearings, crank cheeks and crank pins, with an outlet at each bearing and crank pin, so that with oil under pressure connected to the end of the crank shaft all bearings are flooded with oil and the wrist pin receives its lubrication from the crank pin through a hole in the connecting rod, provided with a check rod running the entire length of the hollow connecting rod. When the engine is in operation this rod plays about one-eight of an inch between the wrist pin and the crank box for the purpose of checking the oil which has once passed the rod, and retaining same for wrist-pin lubrication. With this pressure system there is really no need for a special check rod for the oil in the connecting rod, but this provision

is of considerable advantage, inasmuch as there is no oil in the rod when the engine has been standing for a sufficient time to enable the oil to leak out. With the oil in the rod the wrist pin will get its lubrication right from the start, and this prevents any cutting of the bronze bushing which bears against the steel-hardened and ground wrist pin. In the customary way of putting the wrist pin directly through the piston, usually cast with heavy bosses on each side, the chances of conducting heat to the wrist pin are much greater than in the wrist-pin carrier arrangement used in this engine.

Particular attention is drawn to the illustration showing the conical piston with its even thickness of metal, to the connecting rod, to the piston-pin carrier and to the piston pin, where it will be noted that the wrist pin is mounted in a separate carrier which is locked inside of the piston by means of a snap ring, thus dispensing with the ordinary piston construction calling for heavy bosses on the body of the piston. Inasmuch as there is less heat conducted with the arrangement here shown, the lubrication of the wrist pin and durability of it is materially increased. The heat flowing from the piston wall to the wrist pin in the other style of engine necessarily makes the lubricating oil very thin, and when the engine is shut down and the lubricating-oil feeds ceased, the pin becomes practically dry. Later, when the engine is started, there is a very good chance of trouble, because it takes several minutes before the oil can reach the pin. In the design of wrist-pin carrier here set forth, the temperature is lowered and, with a check rod in the connecting rod, the lubricating oil is kept at a level which will provide lubrication for the pin immediately with the starting of the engine.—"Marine Engineering."

## ELECTRIC PROPELLING MACHINERY FOR THE BATTLESHIP *TENNESSEE.*

While electric propulsion gives approximately the same overall economy in steam consumption as a gear-turbine drive, nevertheless it offers an added advantage of greater latitude in the location of the propelling machinery, a feature which is of greatest importance in the design of naval vessels. The United States Navy has taken the lead in the application of electric drive to battleships and battle cruisers; and of the recent installations, that of the battleship *Tennessee* is of special interest, as it involves certain features which have been developed as a result of previous installations of electric propelling machinery in vessels of this class. The principal details of the installation on the *Tennessee* were described by Wilfred Sykes, general engineer for the Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa., in a recent issue of "The Electric Journal," from which the following information is taken:

The *Tennessee* will be one of the most powerful fighting ships built, having displacement of over 32,000 tons and a speed of 21 knots at full power. The equipment for propelling the ship will consist of four 3-phase induction motors, each driving one propeller, and two turbo generators for supplying the motors with power.

Each of the four motors will develop 7,000 horsepower at a speed of about 180 revolutions per minute, and will be capable of working continuously at 8,375 horsepower as an overload condition. The motors have two windings of 24 and 36 poles, so that they have two normal speeds of 123 and 180 revolutions per minute with full speed of the turbine. In this way it is possible to run the turbine at its most economical speed when steaming either at full power or cruising at 15 knots. Intermediate speeds are obtained by varying the speed of the turbine, and the equipment is

designed to maintain a low water rate over the full range of speed from ten knots up. When operating below 17 knots, only one generator is used, and this improves the economy, as the load on the unit is brought nearer to its full capacity.

### TWO-POLE GENERATORS.

The turbo generators supplying power to the main units each develop 13,500 k. v. a. at full speed and are capable of carrying 15,000 k. v. a. continuously for the overload conditions. The generators are two-pole machines and the unit runs at 2,190 revolutions, corresponding to 36.5 cycles per second, with the motors running at 180 revolutions per minute. The maximum speed of the turbo generator is 2,270 revolutions per minute, corresponding to 37.9 cycles, equivalent to a motor speed of 186.5 revolutions per minute, which requires 8,375 horsepower. To obtain lower speeds, the turbine speed is reduced to about 1,500 revolutions per minute, which corresponds to the change-over point from the 24 to 36-pole connection of the motors.

With the change-over of the motor connections, the speed of the generator is increased to 2,270 revolutions, corresponding to 15 knots with the 36-pole connection. The motor speed combination is simply the equivalent of a variable ratio of gearing, which, in the case of the 24-pole connection, is 12 : 1, and with the 36-pole connection 18 : 1.

The direction of rotation of the machines is controlled by reversing switches which simply transpose two of the phases of the motors, the generator, of course, continuing to run in the same direction.

The motors have two separate windings on the stator, one the 24-pole and the other the 36-pole connection. The same results might have been obtained by use of one winding, but this would have entailed greater complication in the connections and would have restricted the design in other ways.

The rotor has a single polar winding for 24 poles, which is connected to the slip rings in the ordinary way, so that resistance can be inserted in the circuit during starting or reversing. When the machine is operating on the 36-pole connection the rotor winding cross-connections act as short-circuiting connections for this pole combination. With the 24-pole combination, they act as equalizing connections between points of equal potential. On the 36-pole connection the motor operates as a squirrel-cage machine, and it is not intended that this winding should be used during the starting or reversing, but only as a running winding. In this way only one winding is used and one set of slip rings.

### THE TURBINE GOVERNOR.

The speed of the turbine is varied by means of a unique hydraulically-operated governor. The loading of the governor is regulated by means of a variable-pressure oil system, the pressure of which is regulated with great accuracy by a pressure regulating mechanism operated by the control handle. In this way any mechanical connection through shafts or rods with the governor from the operating point, with the consequent danger of jamming where passing bulkheads, is avoided. A unique feature of this arrangement is that the pressure is caused to pulsate slightly so that the whole of the regulating and governor mechanism is kept slightly in motion, and thereby prevented from sticking, also adding greatly to the sensitiveness of the control, which is of great importance when ships are steaming in formation.

The turbines are of the Westinghouse semi-double flow, impulse-reaction type. The high pressure steam is expanded in suitable nozzles and

passes through a two-row impulse wheel, after which it passes through the first stage of the reaction expansion, which is single flow. The steam then divides and passes through the low-pressure stages of the turbine, which are double flow. The turbine is provided with an automatic stop to cut off steam in case the speed should exceed the maximum safe operating value. The main hydraulically operated governor maintains speed practically constant at any value set by the control mechanism, independent of the load, so that in case the propellers should leave the water during rough weather there will be no racing.

### INSULATION.

The generators and motors are very carefully insulated for this service, so as to prevent damage due to moisture or the accumulation of salt, and also due to the high temperatures which are liable to be encountered in this service. The principal material used for insulation of coils in the slots is mica, and the machines are capable of withstanding slot temperatures up to at least 150 degrees C. without injury. Ventilation of the generators is provided by fans supplying air to each engine room and the fans on the generator forcing the air through the machine and out through ducts. The motors each have two fans mounted directly above them which draw the air through the motor and force it out through the ventilating ducts. The generators are excited from the direct-current power circuit of the ship through boosters which are capable of raising the normal 240-volt supply to 320 volts or reducing it to zero.

The power supply from the turbo generators is brought to a centrally-located control room in which is mounted all the necessary switching apparatus for controlling and distributing the power to the motors. In this room is mounted the regulating apparatus for the main turbines, the field switch and rheostat for the turbo generator excitation, and the liquid rheostats for the main motors. All necessary instruments for the operation of the equipment are mounted directly in front of the operators and full advantage is taken of the great facility with which electric power can be measured, inasmuch as it will assist in the operation of the ship.

### AUTOMATIC LIQUID RHEOSTATS.

For starting and reversing the main motors, automatic liquid rheostats are used which are of a similar design to those used previously for industrial purposes. These liquid rheostats consist of two tanks, the upper containing a series of fixed electrodes and the lower acting as a reservoir. By means of a suitable pump, the electrolyte is caused to flow from the lower to the upper tank at the proper rate to cause the desired acceleration. When the by-pass between the two tanks is open the electrolyte is maintained within the electrode tank at the proper level to give the maximum resistance. When this by-pass is closed the electrolyte rises in the upper tank, thereby progressively short-circuiting the electrodes until the minimum resistance is reached at the overflow point, after which the liquid simply continues to circulate through the two tanks. A switch is provided so that this rheostat may be short-circuited.

The cables for connecting the turbo generators and motors are of great importance, as the operation of the ship depends upon their reliability. A number of parallel circuits are used, each cable being of the three-core type, and the failure of any single cable would not seriously interfere with the operation of the ship. The main cables are of the same order of importance as the main steam pipes; hence the greatest care has been taken to insure that these cables should be the best type that is possible

to manufacture for the service, and to this end a committee of the American Institute of Electrical Engineers assisted the Navy Department in preparing the specifications.

ENGINE-ROOM AUXILIARIES.

The main auxiliaries in the engine room are electrically driven. The main circulating pumps are driven by 235-horsepower, direct-current motors, directly connected to centrifugal pumps, the speed of which can be varied to suit various conditions of operation. In this way the power consumption can be reduced as the speed of the ship is reduced. The principal provision for maintaining vacuum in the condenser is the use of LeBlanc air ejectors, which are novel for this class of ship. These air ejectors have already been successfully tried out by the Navy Department on other vessels with such satisfactory results as to justify their adoption for these vessels. The condensate from the condensers is handled by vertical electrically-driven centrifugal condensate pumps, so that the whole of the essential auxiliary apparatus for the turbines is rotary, and, based on the experience in land service as well as at sea, a greater reliability and lower maintenance can be anticipated for these equipments compared with past practice. The use of air ejectors enables the vacuum to be maintained at least as high, if not higher, than the older combination of reciprocating air pump and Parsons augmentor, and the space and weight are a very small fraction of that required with the older system.

While steam consumption is not of vital importance, on account of the other advantages of electric drive, yet the figures that can be obtained are very appreciably better than the direct-connected turbine and are lower than any past practice. From 10 to 15 knots only one generator is used, the motors being connected for 36 poles. From 15 to 17 knots the motors are connected for 24 poles and one generator is used. From 17 knots to 21 knots both generators are in operation, each machine supplying power to two motors, each side of the ship being independently operated.

In designing the equipment for the *Tennessee*, every effort has been made to avoid the introduction of experimental or risky constructions, and the design is such that the experience gained in other fields has been utilized to full advantage, and the design factors have been kept within well-developed practice. At the same time, provisions have been made so that the full advantages of the characteristics of electric drive can be utilized in the operation of the ship.—"Marine Engineering."

## SEVEN WORKSHOP HINTS.

By Fred Horner.

*Repairing Broken Hack-saws.*—Fig. 1 shows a method the writer has found very effective in dealing with hard hack-saw blades which have broken, and need a fresh attachment in a shortened frame. It is difficult and time wasting to attempt softening and drilling for a fresh hole, and a far quicker dodge is to run a groove into the saw with a narrow grinding wheel. This takes but a few seconds, and the heat developed softens the blade in the area, so that it is better able to resist breakage than if hard. The slot is then hooked over the pin and the frame strained up as usual.

*Soft Jaws for Pliers.*—When soft brass or other material has to be bent with the pliers, the hardened serrated jaws of the latter damage the surface of the metal. Fig. 2 shows how to fit readily detachable clams,

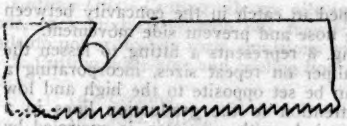


FIG. 1.

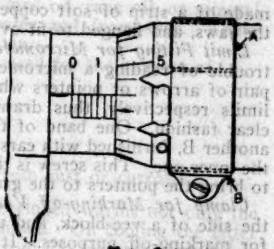



FIG. 3.

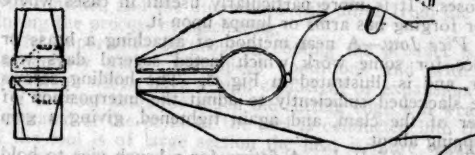


FIG. 2.

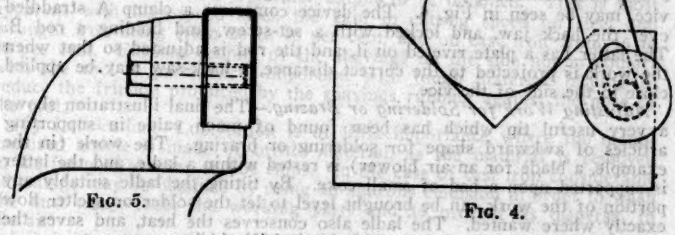


FIG. 5.

FIG. 4.

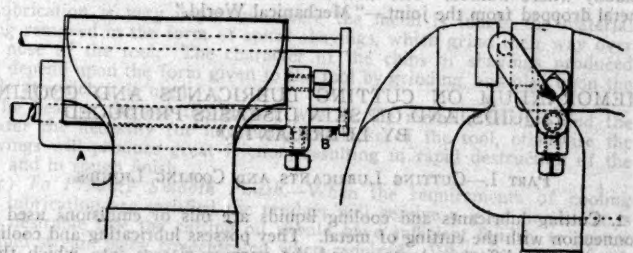



FIG. 6.

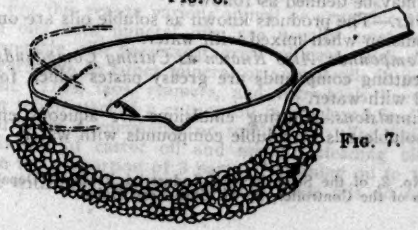


FIG. 7.

made of a strip of soft copper, looped to catch in the concavity between the jaws, and flanged to fit over the nose and prevent side movement.

*Limit Fitting for Micrometer.*—Fig. 3 represents a fitting to lessen the trouble of reading a micrometer caliper on repeat sizes, incorporating a pair of arrows or pointers which can be set opposite to the high and low limits respectively, thus drawing attention to the graduation lines in a clear fashion. One band of thin metal A, with a pointer, is encircled by another B, furnished with ears and screw and nut to draw it tightly around the inner one. This screw is tightened after twisting the bands respectively to bring the pointers to the graduations desired.

*Clamp for Marking-off Vee-block.*—Fig. 4 shows a clamp attached to the side of a vee-block, and used to bind a shaft with sufficient firmness for marking-off purposes. It is more particularly useful in cases where a shaft or a casting or forging has arms or lumps upon it.

*Attaching Clam to Vice Jaw.*—A neat method of attaching a brass or copper clam to a vice for some work which lasted several days was noticed by the writer, and is illustrated in Fig. 5. The holding screws of the steel jaws are slackened sufficiently to admit the interposition of the bent round corner of the clam, and again tightened, giving a grip that permits of no slipping about.

*Stop for Vice for Cutting Off Rods.*—A fitting for a bench vice to hold a stop-rod, without any necessity for drilling or otherwise mutilating the vice, may be seen in Fig. 6. The device comprises a clamp A straddled over the back jaw, and locked with a set-screw, and holding a rod B. The latter has a plate riveted on it, and the rod is adjusted so that when the work is projected to the correct distance, a hack-saw may be applied close to the side of the vice.

*Adjusting Work for Soldering or Brazing.*—The final illustration shows a very useful tip which has been found of much value in supporting articles of awkward shape for soldering or brazing. The work (in the example, a blade for an air blower) is rested within a ladle, and the latter is supported upon a bed of small coke. By tilting the ladle suitably any portion of the work can be brought level to let the solder or spelter flow exactly where wanted. The ladle also conserves the heat, and saves the metal dropped from the joint.—"Mechanical World."

---

## MEMORANDUM ON CUTTING LUBRICANTS AND COOLING LIQUIDS, AND ON SKIN DISEASES PRODUCED BY LUBRICANTS.*

### Part I.—Cutting Lubricants and Cooling Liquids.

1. Cutting lubricants and cooling liquids are oils or emulsions used in connection with the cutting of metal. They possess lubricating and cooling properties in different degrees, and the various classes into which they are divided may be defined as follows:

*Soluble Oils.*—The products known as soluble oils are only liquids which form an emulsion when mixed with water.

*Soluble Compounds, Also Known as Cutting Compounds.*—Soluble compounds or cutting compounds are greasy pastes which form an emulsion when mixed with water.

*Cutting Emulsions.*—Cutting emulsions are aqueous emulsions formed by mixing soluble oils or soluble compounds with water.

---

*Bulletin No. 2, of the Scientific and Industrial Research Department. Reprinted by permission of the Controller, H. M. Stationery Office.

*Cutting Oils.*—Cutting oils are oils such as lard oil, rape oil, or mineral oils, or a mixture of such oils free from water and from soap. These oils do not ordinarily form emulsions with water.

2. Cutting lubricants and cooling liquids are used for the purpose of—

(*a*) Cooling.
(*b*) Lubrication.
(*c*) To produce smooth finish.
(*d*) To wash away chips.
(*e*) To protect the finished product from rust or corrosion.

(*a*) *Cooling.*—During operation the heat developed warms not only the tool, but also the material which is being machined. On cooling the latter will contract, and the dimensions will differ from the measurements taken during the process of machining.

The importance of properly cooling the product is, therefore, obvious, particularly under high-speed conditions and with materials, such as aluminum, which have a high coefficient of expansion.

If the tool heats too much the cutting edge will wear rapidly. The heat generated at the point of the tool is conducted into the body of the tool; if the tool is of large section the heat is more readily dissipated than is the case with a tool of light section. Efficient cooling of the tool edge reduces wear and enables a greater output to be obtained. This is most apparent with high-speed steel, the gain in cutting speed on steel and wrought iron being from 30 to 40 per cent, and on cast iron from 16 to 20 per cent. Efficient cooling of the shavings on the side not in contact with the tool is particularly important with tough material, helping to reduce the friction produced by the shavings rubbing against the nose of the tool.

(*b*) *Lubrication.*—Lubrication is of little importance where the manufactured article is made of brittle material, as the material is removed in the form of powder or fine chips. In the case of cast iron considerable advantage may be obtained by using an aqueous emulsion in order to wash the dust away from the working parts and to prevent its dispersal in the air.

Lubrication is very important where the metal is tough, the material being removed in the form of spiral shavings, which grind their way over the nose of the tool. The character of the chips or shavings produced will depend upon the form given to the tool by grinding, and also upon the angle at which it is used.

The heavier the cut the greater will be the metallic friction and the greater the necessity for lubricating the nose of the tool, otherwise the shavings will produce great friction, resulting in rapid destruction of the tool and in rough finish.

(*c*) *To Produce Smooth Finish.*—When the requirements of cooling and lubrication are satisfied the product will receive a good finish. It is also desirable that the cutting oil should have sufficient fluidity to ensure a rapid stream being concentrated when required. Where a perfect finish is desired, experience has shown that cutting oils possessing great oiliness must be applied. For this reason various animal or vegetable oils, or rich mixtures of such oils with mineral oils, are usually employed. Some engineers find vegetable oils possessing great oiliness, such as rape or cotton seed oil, preferable to either mineral or animal oils in producing a very smooth finish. Dies, taps, reamers, and form-tools have a longer life when used on tough steel if a cutting oil is employed in place of an emulsion prepared from a compound or soluble oil. For finish boring, rifling, etc., a mixture of castor oil and mineral cleaning oil (gravity about 860—890) in the proportion of 3 parts of cleaning oil to 1 of castor oil has been used with good results; although those oils do not form a

homogeneous mixture, the addition of an equal volume of turpentine substitute (white spirit) causes perfect solution to take place, and is said to be advantageous for finish-turning on guns and hard material.

(*d*) *To Wash Away Chips.*—Frequently the washing away of chips is quite an important function of the cutting lubricant or cooling liquid, particularly in cases of deep drilling, as in drilling rifle barrels and the like, also in most milling operations.

If the cutting emulsion is used too weak it will not carry away with it the minute particles of metal and scale which may prove detrimental to the machine tool.

In the boring of deep holes, gun tubes, etc., a solution of sodium carbonate (50 pounds) and soft soap (25 pounds) in water (200 gallons) has been found to give very satisfactory results.

In solid deep-hole boring, where cutting emulsions are used, it is sometimes found that the emulsion, in filtering through the chips in the bore, becomes changed in character in such a manner as to lose some of its lubricating quality.

(*e*) *To Protect Finished Product from Rust and Corrosion.*—Good cutting oils used "straight" (*i.e.*, not emulsified with water) will not cause rusting.

Cutting oils containing fixed oils (animal or vegetable oils) such as lard oil, with a large percentage of free fatty acid, will cause verdigris on brass parts. Vegetable oils containing only a small percentage of free fatty acid, such as rape oil, when employed in cutting oils do not produce verdigris unless the oils are rancid.

Cutting emulsions made up from cutting compounds or soluble oils and water cause rusting if they are used too weak, or if they contain acid. Water from some sources contains considerable quantities of sodium chloride (common salt), which is most destructive to the emulsion; in such cases a supply of water free from salt must be used.

Emulsions of oil and water are not stable in the presence of even minute quantities of acid. The acid causes separation of the emulsion into layers of oil and water, the water is circulated by the pump and causes rusting of the work. To a limited extent the emulsion can be reformed by adding a calculated quantity of ammonia sufficient to neutralize the acid, but any excess of alkali may facilitate corrosion of the metal being worked.

The admixture with a soluble oil of paraffin oil (5 per cent or more) prior to the addition of water has been reported to give good results; a thin film of paraffin oil forms on the top of all standing oil in barrels and tanks and prevents the access of air; similarly a thin film of paraffin oil forms over machined parts, machines and tools, which prevents gumming and rust. It should be noted, however, that the addition of paraffin to a soluble oil reduces its lubricating and emulsifying properties.

Emulsions must not be made by mixing soluble oils or cutting compounds with hard water owing to the precipitate caused by the action of the calcium and magnesium salts in such water. Soft water must be used, which may be either rain water or distilled water, or, if neither of these is available, the hard water must be softened by chemical means.

SELECTION OF CUTTING LUBRICANTS.

3. Before selecting the correct grade of cutting lubricants it is necessary to consider several important factors such as—

(*a*) Cutting speed and depth of cut.
(*b*) The material under manufacture.
(*c*) The system of application of the lubricant or emulsion.
(*d*) The production of skin diseases.

(*a*) *Cutting Speed and Depth of Cut.*

*Low Speed and Shallow Cut.*—Low speed and shallow cut requires little cooling and little lubrication.

*Low Speed and Heavy Cut.*—Low speed and heavy cut, particularly if the material is tough, demands a cutting lubricant possessing great oiliness.

*High Speed and Shallow Cut.*—High speed and shallow cut demands a cutting medium with great cooling properties, consequently emulsions are frequently used. Where a perfect finish is desired, low viscosity cutting oils are used "straight."

Where the speeds are particularly high emulsions only should be used, as otherwise there will be excessive heating of the tools and of the product.

Turpentine substitute ("white spirit") is a satisfactory lubricant for aluminum, but it possesses the undesirable quality of inflammability, and if employed must be used with care on this account.

A mixture of petroleum burning oil (paraffin oil) with lard oil or other cutting oil for high-speed work in connection with aluminum is also dangerous and has led to several fires. It is better to use cutting emulsions which possess the necessary cooling properties and are not inflammable.

*High Speed and Heavy Cut.*—High speed and heavy cut demands a cutting lubricant with great cooling as well as lubricating properties, so that heavily compounded cutting lubricants of low viscosity must be used. Low viscosity is necessary to give good cooling effect, and heavy compounding with animal or vegetable oils is requisite so as to lubricate the tools and shavings effectively and prevent wear as far as possible. For tough material and heavy cuts an emulsion containing 15 to 20 per cent of a vegetable oil has been reported to be satisfactory.

(*b*) *Material Under Manufacture.*—The influence of the material upon the choice of cutting oil has already been referred to (see Section 2).

Where material is brittle, cutting emulsions are nearly always used.

Where the material is tough, but where the speeds are high and the cut light, cutting emulsions are also frequently employed; but where the material is tough and the cut heavy it is necessary to employ cutting lubricants used "straight" and containing a percentage of animal or vegetable oils (ranging from 10 to 50 per cent) or consisting entirely of such oils.

With automatic screw-cutting machines, emulsions have been found in some cases to form a deposit in the working parts of the machines. This may be avoided by the employment of "straight" oils, but in view of the present scarcity of oil, it is advisable in the national interest to use emulsions wherever possible.

The amount of soluble oil or soluble compound used for preparing the cutting emulsion varies from 2½ to 20 per cent, the richer mixtures being used for severe conditions, and the weaker mixtures for light duties or for materials such as brass and aluminum where there is no danger of rusting.

(*c*) *System of Application.*

The cutting lubricant may be applied by hand by a drop-feed system or by some system employing gravity or pumping.

In large machine shops the cutting oil is sometimes circulated through pipes throughout the works, returning through other pipes from the machine tools to a central tank.

Group systems with central tanks are excellent where one mixture is used for all machines on the circuit.

Return pipes should be large, and should be arranged for easy access for cleaning; in large systems isolating valves should be employed to sectionize the system; efficient strainers should be fitted on all return pipes and pump sections and should be cleaned daily. Tanks as a rule should

be cleaned out every four weeks and return pipes every four months. Any scum formed should be skimmed off the tanks daily.

It is important both with this system and where machine tools have individual pumps that the pump's suction should be always covered so that air cannot be drawn into circulation, since aeration of the circulating medium has a strong oxidizing effect upon the oil or emulsion. The important point to keep in mind in regard to the system of application of the lubricant is that where the oil or emulsion is circulated over and over again, it is exposed to the oxidizing effect of air and to admixture with dust and dirt from the machine shop.

A certain amount of oil or emulsion is always splashed away from the machine tools in the form of spray, and is lost notwithstanding precautions taken by the use of splash guards; the loss of oil depends very largely upon the viscosity of the cutting medium.

## PHYSICAL AND CHEMICAL POINTS OF INTEREST.

4. *Cutting Oils.*—The mineral oils, which are best suited to be used as cutting lubricants, either alone or mixed with animal or vegetable oil, are mineral oils, preferably of pale color, of low viscosity, ranging from 100 secs. to 200 secs. Redwood at 100 degrees F. The lower viscosity oils may be used for high-speed conditions, and oils with higher viscosity may be used for slow-speed conditions. Of the animal oils used either alone or in admixture, tinged lard oil containing as much as 10 to 15 per cent of free fatty acid is most frequently employed. Prime lard oil is almost free from acid; it is much more expensive than tinged lard oil, but is less inclined to gum under severe conditions (heavy cut and high speed).

Lard oil congeals in cold weather, so that, wherever possible, a mixture of lard oil and low cold-test mineral oil is to be preferred on account of greater fluidity in the cold. Experience has proved that most cutting lubricants containing vegetable oils, particularly if they are heavily blown (*i.e.*, thickened by oxidation), are liable to produce gummy deposits in circulation systems. These deposits interfere with the proper operation of the machine, and necessitate frequent cleaning of the machines, which not only increases the costs, but also decreases the output.

Cottonseed oil oxidizes more readily than rape oil, and should not be used in the manufacture of cutting lubricants that are to be used in a circulation system. Animal oils are not so easily oxidized in a circulation system as are vegetable oils.

Experience shows that lard oil only produces deposits in circulation systems under severe operative conditions when the percentage of free fatty acids exceeds, say, 10 per cent.

When oils of low volatility are used a certain proportion of the oil is evaporated by the heat produced in the work; this is objectionable on account of the smoke and fumes created. Where cutting emulsions are used steam only is produced.

Cutting oils are nearly always used "straight"—*i.e.*, without admixture of water.

Certain cutting oils containing at least 5 per cent of free fatty acid and preferably more than 20 per cent of saponifiable oil (animal or vegetable oil) may either be used "straight" or in the form of cutting emulsions. They will emulsify with water to which the requisite amount of alkali (soda ash, borax, etc.) has been added.

*Soluble Oils and Soluble Compounds.*—Soluble oils are prepared by dissolving a soap (usually less than 20 per cent) in a mixture of mineral oil (usually less than 70 per cent) and saponifiable oil (usually more than 15 per cent).

The saponifiable oils used in making the soap are either of animal or vegetable origin, such as lard oil or other olein from animal fat, whale oil, wool grease, castor oil, sulphonated castor oil, rape oil, cottonseed oil, resin, resin oil, etc., and the oils are saponified by means of caustic soda or caustic potash.

Some soluble oils or compounds contain a small percentage of alcohol or ammonia, which causes the emulsion to form more readily.

When soluble oils containing ammonia are stored for several months in wooden barrels, the ammonia is sometimes absorbed by the barrel, and the oil is no longer completely soluble in water.

With cutting emulsions made from soluble oils containing ammonia, the ammonia volatilizes under severe conditions of service, and a scum is produced on the surface of the emulsion which is objectionable, as it tends to clog the pipes in the circulation system, and by adhering to the swarf or chips causes considerable loss of oil. Soluble oils containing a large percentage of resin or resin oil have a tendency to cause gumming.

Soluble compounds are made on similar lines to soluble oils, except that they contain 10 to 50 per cent of water and are in a semi-solid and semi-emulsified condition. They are not so easily mixed with water as are soluble oils, and for this reason the latter are usually preferred.

PART II.—SKIN DISEASES PRODUCED BY LUBRICANTS.

1. Oil rashes are, generally speaking, of two kinds—the first is due to plugging of the small glands at the root of the hairs on the arms and legs of workers, the second to mechanical injury to the skin produced by metallic particles suspended in the cutting lubricant.

(*a*) *Plugging of the Glands of the Hair Follicles.*

Primarily this is purely mechanical; a mixture of oil and dirt blocks the minute openings of these glands and sets up inflammation round the hair (folliculitis). The inflammation commenced in this way may lead on to suppuration or abscess formation (a boil). If many hairs are effected the arm presents an appearance of a crop of raised red spots (papules), with a black spot as a center, or, if the inflammation has gone as far as suppuration (abscess formation), a yellow head.

(*b*) *Mechanical Injury to the Skin by Metallic Particles.*

Minute metallic particles suspended in the cutting lubricant may produce injury to the skin. This occurs chiefly on the hands, where two surfaces are rubbed together—*e.g.*, the skin between the fingers. Injury to the skin may also be produced on any part of the hands and arms by wiping with a cloth or rag while the hands or arms are coated with a film of fluid in which metallic particles are suspended. Injury to the skin allows germs to enter and causes septic infection.

2. *Prevention.*—(*a*) *Cleanliness of the Worker.*—Washing accommodation for workers in contact with oil must be on a liberal scale. Hot water, soap, and scrubbing brushes are essential. Workers should be instructed not to wipe their hands on rags, etc., before washing, and to avoid washing their hands in the cutting compounds.

Ether soap, which dissolves oil, has been found useful in preventing inflammation of the hair follicles. Dusting the arms with a powder containing equal parts of starch and zinc oxide before commencing work prevents the action of the oil on the skin.

(*b*) *Cleanness of the Lubricant.*—Care must be taken in the handling of the constituents before blending that they have not undergone changes (*e.g.*, formation of free fatty acid).

Constant removal of metal particles is necessary to avoid injury to the

skin. Filtration, such as is provided on the machines, and centrifugal action are insufficient to remove the minute metal particles which may injure the skin. Where cutting oils (straight oils) are used, their viscosity can be diminished by heat sufficiently to allow the particles to sink without affecting their value as lubricants. This operation completely removes all metal particles. In other lubricants where such a procedure is impossible it is necessary constantly to change and renew the cutting lubricants.

(*c*) *Cleanness of the Machines.*—Frequent cleaning of the machines with the removal of all the old lubricant from all parts of the machine is essential.

3. *Addition of Disinfectants or Antiseptics to the Lubricants.*—Various antiseptics, carbolic acid (1 to 2 per cent) being the most common, have been added to the lubricant to prevent rashes, and in the case of cutting emulsions 0.5 per cent of disinfectants soluble in water have been used for this purpose. The results obtained have not been altogether satisfactory, and reliance cannot be placed upon such a method to prevent skin rashes.

4. *Sterilization by Heat.*—It has been suggested to heat the cutting oil to 300 degrees F. for a short period with a view to sterilizing it, as well as to increase its antiseptic or germicidal action.

Laboratory experiments in America have shown that used oil possesses rather marked germicidal effects, and in view of the fact that the used oil becomes heated during use, attempts were made to determine whether heating new oil would also bestow upon it germicidal powers. Apparently, heating does produce such a change, but the temperature required is upwards of 125 degrees C. The actual temperature required to produce this germicidal action in the oil has not yet been determined, but it has been recommended to mix new oil with the used oil before filtering and heating, so that the new oil would possess to some extent the germicidal power of the used oil.

5. *Removal of Workers with Septic Infection of the Hands.*—Workers whose hands become the seat of septic infection should not be allowed to work on machines, as they are liable to infect the oil with germs and so infect others.

6. *Treatment.*—(*a*) *Folliculitis Produced by Blocking of the Glands.*—As a general rule, frequent washing with soap and hot water is sufficient to produce a rapid cure. The skin may be subsequently dusted with zinc oxide and starch powder.

It has been found that where this is insufficient a mild antiseptic applied on lint has relieved the irritation and given good results.

(*b*) *Septic Infection of the Skin Due to Cuts.*—Septic infection should be treated on general principles by the application of suitable antiseptic dressings.

7. *Susceptibility.*—Certain individuals appear to be particularly susceptible to the action of lubricants. Such persons when found should be removed from contact with oil.—"Mechanical World."

# THE PROPULSION OF CARGO SHIPS WITH PARALLEL MIDDLE BODY.

*Variations of Shaft Horsepower, Propeller Revolutions, and Propulsive Coefficient with Longitudinal Position of the Parallel Middle Body in a Single-Screw Cargo Ship.**

By Naval Constructor William McEntee, U. S. N., Member.

Previous experiments made at the United States Experimental Model Basin indicate† that for slow ships of full lines, such as are used for the ordinary single-screw cargo ship, it is desirable to use a certain amount of parallel middle body. In those experiments the different models used had varying lengths of parallel middle body distributed equally forward and abaft the midship section. The present investigation had for its object the determination of the best fore and aft position for the parallel part of the ship, and the investigation of the effect of this variation on the shaft horsepower, propulsive coefficient, and wake and thrust deduction factors.

Four 20-foot models were made corresponding to ships of 400 feet length between perpendiculars and of 57.3 feet beam, 26 feet draught, and 13,137 tons displacement when fully loaded. A longitudinal or prismatic coefficient of 0.788 was chosen as representing about the present practice in cargo carriers of this type. A constant length of the parallel middle body was taken equal to 33 per cent of the ship's length. This percentage was found in the investigations referred to above to give about the minimum residuary resistance for the speeds attained in practice for ships of this type.

In Fig. 1, annexed, are shown the lines of model 2,023 which were used as the parent form. In Fig. 2, are shown the curves of sectional area for the four models and the parent form. Model 2,132 was made very full at the entrance, with a fine run. Model 2,135 was made relatively very fine at the entrance and full in the run. Models 2,133 and 2,134 were intermediate between the two extreme models. The model with the very full bow and that with the very full stern were purposely made of an extreme type beyond anything that might be expected to be used, in order to give a wide scope to the investigation and to insure that the limiting condition as regards power should be obtained at either extreme.

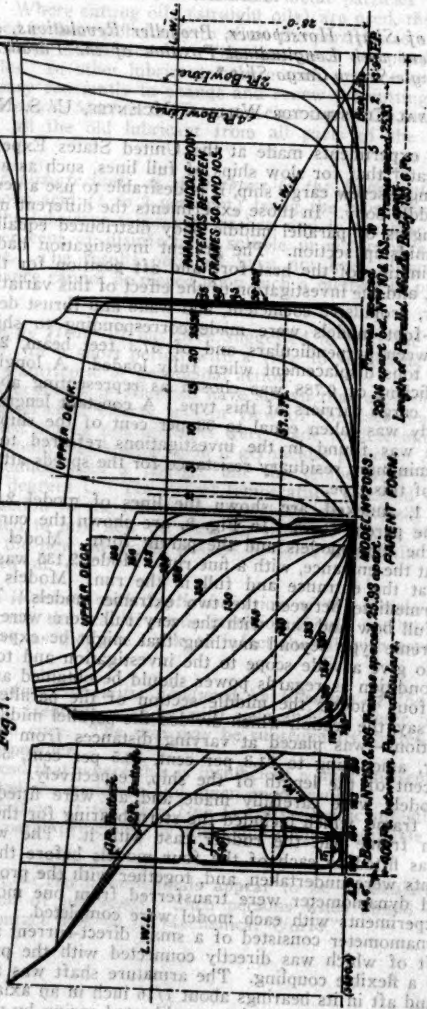
In the four models the middle section of the parallel middle body—that is to say, the section which divided the parallel middle body into two equal portions—was placed at varying distances from the forward perpendicular, amounting to 31.3 per cent, 38.5 per cent, 53.4 per cent and 60.5 per cent of the length of the ship, respectively.

The models were carefully made and all were fitted with the same cast stern frame, which included the stern bearing for the propeller shaft. The stern frame had the rudder cast with it. The whole frame and rudder was fitted to each of the four models before the self-propulsion experiments were undertaken, and, together with the propeller shaft, propeller and dynamometer were transferred from one model to the other as the experiments with each model were completed.

The dynamometer consisted of a small direct-current motor, the armature shaft of which was directly connected with the propeller shaft by means of a flexible coupling. The armature shaft was free to float fore and aft and aft in its bearings about 7/16 inch in an axial direction. The armature shaft was connected to a calibrated spring by means of a thrust

*Paper read before the Society of Naval Architects and Marine Engineers, at Philadelphia, November, 1918.

†"Speed and Power of Ships," by D. W. Taylor.


PARALLEL MIDDLE BODY EXTENDS BETWEEN FRAMES 150 AND 105.
2 P.R. Bow Line
4 P.R. Bow Line
Upper Deck.
Upper Deck.
MODEL N°2085.
PARENT FORM.
Length of Parallel Middle Body 78.0 ft.
Fig. 1.

bearing, so that the axial displacement of the armature shaft gave a measurement of the propeller thrusts. Similarly, the frame of the motor was mounted so as to rotate in independent bearings. The torque developed by the motor acted against a calibrated spring so that the deflection of the spring indicated the torque of the motor. In addition to this there were suitable means provided for measuring the revolutions of the shaft.

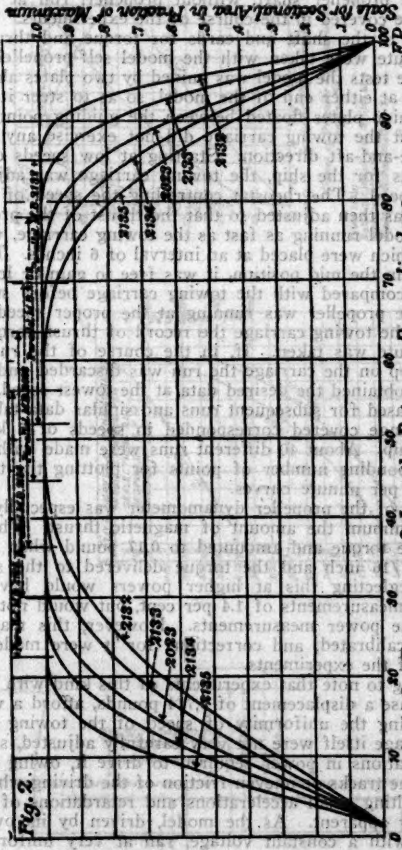



Fig. 2.

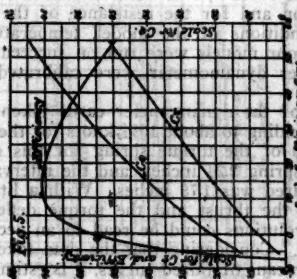
The order of procedure in making the tests was as follows: The shaft and dynamometer were carefully lined up and the whole run for a sufficient time to warm up the bearings and reduce the bearing friction as much as possible. Owing to the fact that the dynamometer was placed

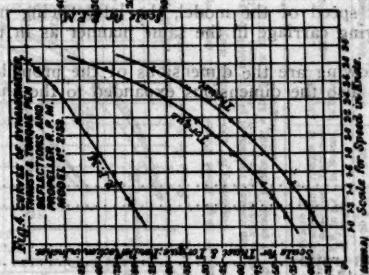
very close to the stern, but a short length of propeller shafting was necessary, and this was supported by two self-aligning bearings, one at the stern bearing and the other at the forward end of the stern tube. With the propeller shaft in place and everything working freely the model was towed in the Model Basin beneath the towing carriage at several different speeds, and the propeller shaft, without propeller, run at the range of revolutions to be covered in the course of the experiments. The propeller was then fitted to the shaft and cards for torque and thrust and revolutions per minute were taken with the model self-propelled at different speeds. In these tests the model was guided by two plates about 10 inches in width placed at either end of the model, so as to steer it in a straight course. The guide plates floated between the guiding points attached to the carriage, but the towing carriage did not exercise any force on the model in a fore-and-aft direction. Starting at low speeds corresponding to about 5 knots for the ship, the towing carriage was adjusted to run at a uniform speed. The rheostat controlling the speed of the propeller dynamometer was then adjusted so that the thrust of the propeller would just keep the model running as fast as the towing carriage, without striking the stops, which were placed at an interval of 6 inches. Thus, starting with the model in the mid position, it was free to gain or lose a distance of 3 inches as compared with the towing carriage before striking either stop. When the propeller was running at the proper speed to keep the model up with the towing carriage the record of thrust, torque and revolutions per minute was taken. If, in the course of the run, the model struck either stop on the carriage the run was discarded and another run made. Having obtained the desired data at the lowest speed, the carriage speed was increased for subsequent runs and similar data taken at higher speeds. The range covered corresponded to speeds of 5 knots to 12½ knots for the ship. About 40 different runs were made with each model, giving a corresponding number of points for plotting the torque, thrust and revolutions per minute curves.

The armature of the propeller dynamometer was especially designed to reduce to a minimum the amount of magnetic thrust. This thrust increased with the torque and amounted to 0.17 pound when the armature was displaced 7/16 inch and the torque delivered to the shaft was 16 pound-inch. Neglecting this at higher powers would have caused an error in thrust measurements of 1.4 per cent, but would not have caused any error in the power measurements. However, this magnetic thrust was separately calibrated, and corrections for it were made in working up the results of the experiments.

It is interesting to note that experiments of this kind with a very heavy model, in this case a displacement of 3,774 pounds, afford a very sensitive means of checking the uniformity of speed of the towing carriage. If the towing carriage itself were not very carefully adjusted, so as to eliminate slight variations in power required to drive it, owing to variations in the level of the tracks or uneven friction of the driving wheels or guide wheels, the resulting small accelerations and retardations of the carriage were made very apparent. As the model, driven by its own propeller, when supplied with a constant voltage, ran at very uniform speeds, it was possible to see the towing carriage gaining or losing distance of a few inches in 5 seconds or 10 seconds, as slight inequalities of speed due to small variations in the resistance of the driving mechanism developed. In order to conduct self-propulsion experiments successfully with the method followed, it was necessary to have the towing carriage running in excellent condition.

Immediately after completion of the self-propulsion tests on the model the propeller was removed and the runs to obtain the shaft friction and





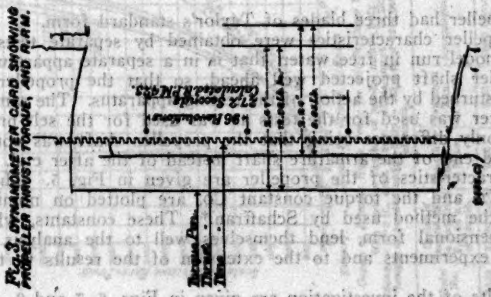



FIG. 8. DYNAMOMETER CARD SHOWING PROPELLER THRUST, TORQUE, AND R.P.M.

the thrust without propeller were repeated. The model was next connected with the resistance dynamometer on the towing carriage and the usual model resistance data taken. This insured that the conditions of test both for self-propulsion and for the resistance of the model would be uniform as regards conditions of the model, temperature of water, etc. Experiments on the four models were run on succeeding days, and the springs of the propeller dynamometer were calibrated both before and after the tests.

In Fig. 3 is shown one of the dynamometer cards taken for a model running at a speed corresponding to about 10½ knots for the ship. It will be noted that the deflection of the torque spring for this condition was 2.47 inches, for the thrust spring 2.24 inches, and the interval over which the revolutions were measured was 1.78 inches. With a little practice it was possible to read both the thrust and the torque deflections within 1/100 of an inch. The revolutions could be read to an accuracy of less than one-half of 1 per cent. Considering that the data taken are plotted and the whole averaged by means of faired curves, it is estimated that the results obtained as regards the three elements measured are correct within 1 per cent.

In Fig. 4 are shown the actual observations of the torque and thrust pen deflections, and the revolutions per minute of the propeller shaft plotted on speed of the model, the latter being measured by the speed of the towing carriage in the same manner as in the ordinary resistance tests.

The following are the dimensions of the propeller used in the experiments and also the dimensions expanded to the ship scale:

| | Model. In. | Ship. Ft. In. |
|---|---|---|
| Diameter | 10.125 | 16 7 |
| Pitch | 9.0 | 14 9 |
| Pitch ratio | 0.889 | |
| Mean width ratio | 0.20 | |
| Number of blades | 3. | |
| Ratio of projected to disc area | 0.266 | |
| Blade thickness fraction | 0.04 | |

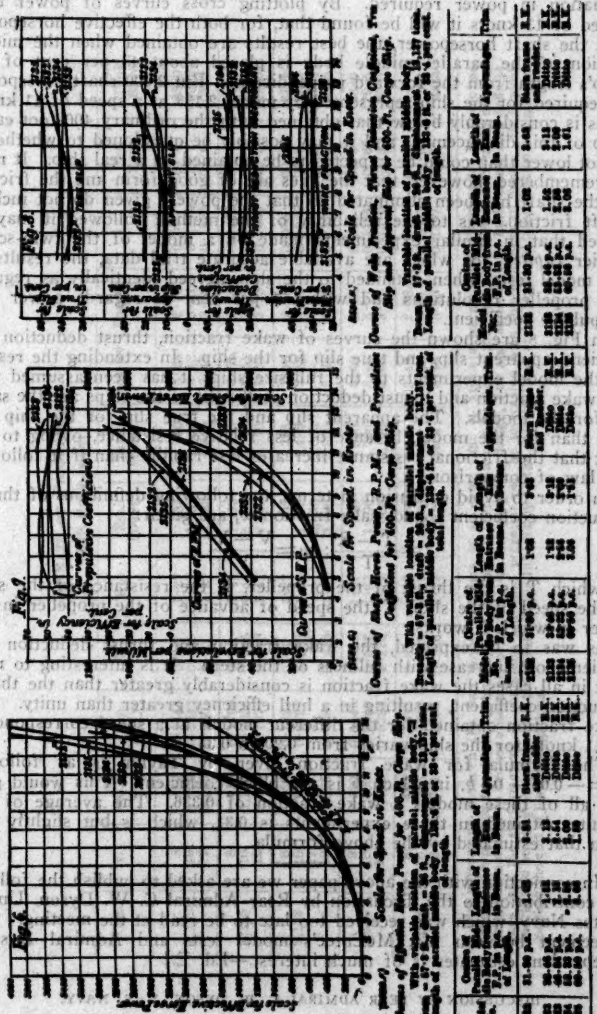
The propeller had three blades of Taylor's standard form.

The propeller characteristics were obtained by separate tests of the propeller model run in free water, that is in a separate apparatus where the propeller shaft projected well ahead, so that the propeller ran in water undisturbed by the action of the testing apparatus. The same motor dynamometer was used for the tests as was used for the self-propulsion tests, the only difference being that the propeller shaft was coupled to the forward end of the armature shaft instead of the after end.

The characteristics of the propeller are given in Fig. 5. The thrust constant $C_T$, and the torque constant $C_Q$, are plotted on nominal slip following the method used by Schaffran.* These constants, which are in non-dimensional form, lend themselves well to the analysis of self propulsion experiments and to the extension of the results to the full-size ship.

The results of the investigation are given in Figs. 6, 7 and 8. An examination of the effective horsepower curves and the shaft horsepower

* "Systematische Propellerversuche," K. Schaffran, Schiffbau, September 22, 1915.



curves for the different models shows, as would be expected, a wide variation in power required. By plotting cross curves of power at a speed of 11 knots it will be found that, for both the effective horsepower and the shaft horsepower, the best results are obtained when the middle section of the parallel middle body is placed about 43 per cent of the ship's length from the forward perpendicular. But 2,000 shaft horsepower is required for the ship represented by model 2,133 at a speed of 11 knots. This is considerably below that obtained with the ordinary 400-foot cargo ship of this displacement and it may possibly be questioned to whether it is not lower than could be expected to be obtained on a real ship. It must be remembered, however, that the lines are of good form and the friction of the shaft has been eliminated so that the powers given do not include shaft friction. As to the reliability of the method followed it may be stated that in similar experiments made on a model of the twin-screw collier *Jupiter*, for which are available accurate trial data, the results of the model tests, when extended to the ship, agreed identically as regards the propeller revolutions and within 1 per cent as regards power and propulsive coefficient.

In Fig. 8 are shown the curves of wake fraction, thrust deduction coefficient, apparent slip, and true slip for the ship. In extending the results of the model experiments to the full-size ships it has been assumed that the wake fraction and thrust deduction coefficient of the ships are the same as for the models. The apparent slip and the true slip for the ship are less than for the model, because of less relative resistance, owing to the fact that the frictional resistance increases less rapidly than if it followed the law of comparison.

In order to avoid confusion in terms, the following definitions of thrust deduction coefficient $t$, and wake fraction $w$, are given:

$$t = \frac{T - R}{T}; \quad w = \frac{V - V'}{V};$$

in which T is the thrust of the propeller, R the resistance of the ship, V the speed of the ship, V' the speed of advance of the propeller in the water in which it works.

As was to be expected, the wake fraction and thrust deduction coefficient both increase with fullness of the stern. It is interesting to note that in all cases the wake fraction is considerably greater than the thrust deduction coefficient, resulting in a hull efficiency greater than unity. The wake fraction obtained for the different models at a speed corresponding to 11 knots for the ship varies from 0.29 to 0.35.

The formula for wake fraction given by Taylor is as follows: $w = -0.05 + 0.5\,b$, in which $b$ is the block coefficient. This would give for all of these models a wake fraction of 0.336. The average of the results obtained in these experiments is 0.31, which is but slightly less than that estimated by the above formula.

---

[In connection with the above paper we are asked to publish the following contribution to the discussion by Rear Admiral C. W. Dyson, United States Navy, which was received too late to be read at the meeting. The agreement between Mr. McEntee's model tests and Admiral Dyson's independent estimates is of much interest.—Ed. E.]

DISCUSSION BY REAR ADMIRAL C. W. DYSON, U. S. NAVY.

Commander McEntee has requested me to analyze the propeller which he used in his experiments and to make an estimate of its performance behind the various hulls which he has used in order to ascertain the degree of agreement between the results obtained by the two methods.

**Estimate of Performance.**

**Shaft Horse-power.**

| Hull. | v. | e.h.p. | $\frac{e.h.p.}{E.H.P.}$ | Z. | K. | Est. Power. | Tank Power. | Log Av | Log As | s. | Est. Revs. | Tank Revs. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 2,132 | 8 | 480 | 0·1415 | 0·89 | 1·27 | 735 | 750 | 3·48 | 2·78 | 0·09 | 60·4 | 62·0 |
| | 10 | 1,100 | 0·3243 | 0·51 | 1·27 | 1,762 | 1,800 | 3·48 | 3·01 | 0·1274 | 78·7 | 80·5 |
| | 12 | 2,580 | 0·7607 | 0·12 | 1·27 | 4,325 | 4,500 | 3·48 | 3·23 | 0·1885 | 101·6 | 109·0 |
| 2,135 | 8 | 610 | 0·1799 | 0·74 | 1·41 | 1,153 | 1,225 | 3·51 | 2·78 | 0·1379 | 63·75 | 69·0 |
| | 10 | 1,240 | 0·3656 | 0·455 | 1·41 | 2,220 | 2,250 | 3·51 | 3·01 | 0·1564 | 81·44 | 86·0 |
| | 12 | 2,500 | 0·7371 | 0·133 | 1·41 | 4,607 | 4,550 | 3·51 | 3·23 | 0·1955 | 102·5 | 109·0 |
| 2,129 | 8 | 450 | 0·1327 | 0·91 | 1·27 | 702 | 725 | 3·48 | 2·78 | 0·086 | 60·8 | 61·5 |
| | 10 | 925 | 0·2727 | 0·568 | 1·27 | 1,472 | 1,480 | 3·48 | 3·01 | 0·1065 | 76·9 | 77·5 |
| | 11 | — | — | — | — | 2,030 | 2,000 | — | — | — | 88·0 | 87·0 |
| | 12 | 1,825 | 0·5381 | 0·276 | 1·27 | 3,020 | 2,850 | 3·48 | 3·23 | 0·1316 | 94·9 | 93·5 |
| 2,134 | 8 | 490 | 0·1445 | 0·855 | 1·27 | 796 | 850 | 3·48 | 2·78 | 0·096 | 60·9 | 63·0 |
| | 10 | 1,000 | 0·2949 | 0·553 | 1·27 | 1,596 | 1,700 | 3·48 | 3·01 | 0·1154 | 77·88 | 79·5 |
| | 12 | 1,825 | 0·5381 | 0·276 | 1·27 | 3,020 | 3,150 | 3·48 | 3·23 | 0·1316 | 94·9 | 97·5 |
| 2,023 | 8 | 490 | 0·1415 | 0·89 | 1·27 | 785 | | 3·48 | 2·78 | 0·09 | 60·4 | |
| | 10 | 925 | 0·2727 | 0·588 | 1·27 | 1,472 | No | 3·48 | 3·01 | 0·1065 | 76·9 | No |
| | 12 | 1,660 | 0·4895 | 0·326 | 1·27 | 2,679 | data | 3·48 | 3·23 | 0·1167 | 93·3 | data |

The agreement between the two power columns is so close that it appears to me that Commander McBeath has developed a method for obtaining quickly and cheaply all such data as it has taken me years to collect and from the data so obtained, not only to deduce absolutely correct factors for propeller design, but also so to classify hulls along such definite lines that the performances of combined hull and propeller can be forecasted with the minimum of error. He is most heartily to be congratulated upon the results of his labours up to the present time, and I hope and trust that he has not yet reached the end of his investigations.

In order to give a clear understanding it will be necessary for me to give a brief description of the method used by me and which has been obtained by many years' study of the performances of actual propellers driving actual ships over carefully measured courses.

The form of propeller blade selected from which to derive the design or performance factors is that of which the projected area is an oval with the greatest circular width at 0.7 radius of the propeller from the center.

From these performances a series of basic curves of design have been obtained from which the performance of the propeller under these basic conditions can be obtained and the performance of the propeller under any other conditions of performance derived from this basic performance by the application of suitable factors entailed by the changed conditions.

The basic conditions are denoted as follows:

I.H.P. = Basic indicated horsepower.

S.H.P. = I.H.P. $\times$ 0.98 = Basic shaft horsepower.

P.C. = Basic propulsive coefficient with maximum hull efficiency. (Taken for total projected area ratio.)

E.H.P. = I.H.P. $\times$ P.C. = Basic effective (tow-rope) horsepower.

P.A. $\div$ D.A. = Projected area ratio (outside 0.2 diameter) of three-bladed basic propeller.

4/3 P.A. $\div$ D.A. = Projected area ratio of four-bladed screw.

⅔ P.A. $\div$ D.A. = Projected area ratio of two-bladed screw.

1 — S = 1 — Basic apparent slip (for three blades) and for fullness of after body of vessel.

$I.T._D$ = Basic indicated thrust in pounds per square inch of disc area of the propeller.

D = Diameter of propeller in feet.

P = Pitch of propeller in feet.

T.S. = Basic tip speed (three blades) in feet, of propeller.

$v$ = Actual speed of vessel.

e.h.p. = Effective (tow-rope) horsepower for $v$.

The power required to deliver e.h.p. where no cavitation exists is expressed by:

$$I.H.P._d = I.H.P. \times K \div 10s$$

where $s$ depends for its value upon the value $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ and K is the thrust deduction factor.

The revolutions corresponding to the actual conditions of resistance are found by the following equations:

$$s = \text{Apparent slip} = S\,\frac{I.H.P._d \times Av}{I.H.P. \times A_v}$$

and

$$R_d = \frac{v \times 101.33}{P \times (1-s)}$$

where S is the basic apparent slip and Av and $A_v$ are factors depending upon the values of V and $v$.

It should be borne in mind that slight variations in the form of blade have only slight effect upon the efficiency so long as the same projected area is retained, but do have a considerable effect upon revolutions, therefore we should be prepared to find but small differences between the estimated and tank powers if both methods are correct, but considerable variation in the revolutions as the propeller used by Commander McEntee had blades of the Taylor form, while the standard-blade form used by me is an oval.

In determining the block coefficient to use for basic apparent slip, the open water (where the propeller is not covered by the limits of the load

water plane of the ship) correction is applied for hulls 2,132, 2,133 and 2,134, directly to the standard block as ordinarily obtained for hulls of the given length, beam and displacement, while for hull 2,135, which had abnormally full after-body lines, this same correction has been applied to an increased standard block, this block being 0.80 in the first three cases and 0.85 in the latter, the corresponding open-water blocks being 0.532 and 0.61.

The work of analysis and estimate follows, the estimated and tank powers and the estimated and tank revolutions being placed in parallel columns.

HULL CHARACTERISTICS.

L.B.P. = 400 ft.
B = 57.3 ft.
H = 26 ft.
Displ. = 13,137.
Nom. B.C. = 0.7716.
B ÷ L.B.P. = 0.1433.
Stand. B.C. for K = 0.80.
Est. Stand. B.C. for K (Hull 2,135) = 0.85.
D < .7 H .·.
Stand. S.B.C. for V = 0.532.
Stand. S.B.C. for V (Hull 2,135) = 0.61.
K = 1.27, for 2,135 = 1.41.

BASIC CONDITION OF PROPELLER.

| | |
|---|---|
| P.A. ÷ D.A. (three blades) | 0.266 |
| D | 16.583 ft. |
| P | 14.75 ft. |
| T.S. | 5,900 |
| P × R | 1,671. |
| 1 — S (for S.B.C. 0.532) | 0.89 |
| (for S.B.C. 0.61) | 0.90 |
| V | 14.67 |
| | 14.84 |
| $I.T_D$ | 3.1 |
| I.H.P. | 4,880 |
| P.C. | 0.695 |
| E.H.P. | 3,392. |
| S.H.P. | 4,490 |

—"Engineering."

## TWO VERSUS FOUR-CYCLE INTERNAL-COMBUSTION MARINE ENGINES.

By Giovanni Chiesa, Manager of Messrs. Ansaldo San Giorgio's Works at Turin.

The relative superiority of two or four-cycle internal-combustion engines for marine purposes is one of the most debated questions at the present moment from a theoretical as well as from a practical standpoint: thus it forms daily the subject of discussions, lectures and articles in technical reviews. The chief purpose of this article is to coördinate the arguments

which have been alleged for and against both types in their best form of construction, and to endeavor to draw a conclusion after careful consideration of all points of the question.

The advantages which are usually attributed to the two-cycle engine as compared with the four-cycle type may be briefly stated as follow:

(A) The two-cycle engine develops a greater power than the four-cycle with the same number and size of the cylinders and the same number of revolutions. This advantage of the two-cycle type is due to the fact that the four-cycle type gives an impulse for each cylinder every two revolutions, whilst the two-cycle type gives an impulse each revolution; theoretically the two-cycle type should therefore develop, under the same conditions, a power double that of the four-cycle type. In practice, however, the said theoretical limit has never been reached, but at present it may be said that the power developed by a two-cycle engine is 175 per cent to 190 per cent of that of a four-cycle engine, and it may be added that whilst the mean effective pressure in the four-cycle type is about 5 kg. per cm.$^2$ (71 pounds per square inch) that of the two-cycle type is practically of 4.4 kg. to 4.75 kg. per cm.$^2$ (62 to 67 pounds per square inch).

This essential advantage of the two-cycle type brings as a consequence a remarkable reduction of space and weight, which may be approximately calculated in the following manner: As there is no reason that a four-cycle cylinder with its framing and driving gear (assuming the same intensity of stress of the materials) should weigh less than a two-cycle cylinder of the same size, and as the weight can be practically considered to be proportional to the volume swept by the piston, therefore, for the same power and number of revolutions, the cylinder of the two-cycle engine (175 per cent being taken as the power-ratio of the two-cycle to the four-cycle type) has a weight which is 57 per cent of that of the four-cycle engine.

This advantage is somewhat reduced by the fact that the two-cycle engine needs scavenging pumps, and as, according to circumstances and to the different design of the pumps, their weight can be considered as being 8 per cent to 12 per cent of the weight of the cylinders, it results that the weight of the two-cycle type will be 62 per cent to 65 per cent as compared with the weight of the four-cycle. The above figures seem also practically confirmed, though there is always some difficulty in comparing numbers quoted by different constructors, for they do not always state which parts of the equipment of the plant are included or excluded from the figures published. But besides the saving of weight there is also the saving of space, and on this particular point it is preferable to refer the reader to the diagrams, Fig. 1, 2, 3 and 4, which show the two-cycle engines of the ship *Ceara* constructed by the firm of Fiat San Giorgio (now Ansaldo San Giorgio), as compared with four-cycle engines of the two best-known types: Burmeister and Wain, and Werkspoor.

It must be noted that the saving in space by the two-cycle type has also as a consequence a considerable saving in the cost and weight of the engine seat as well as in the dimensions of the engine room, facilitating the supervision and control of the machines.

(B) The turning moment in the two-cycle engine is far more regular (for the same number of cylinders) than in the four-cycle type; Fig. 5 compares the diagrams of the turning moments of a four-cycle six-cylinder engine with that of a two-cycle six and four-cylinder type, illustrating the great difference in the regularity of the two types; the results of even the four-cylinder two-cycle type are far more regular than those of the six-cylinder four-cycle engine.

This advantage of the two-cycle engine is not merely theoretical, but in practice results in a minor intensity of the vibrations of the stern end of the ship, besides a reduction in size and weight of the line of shafting

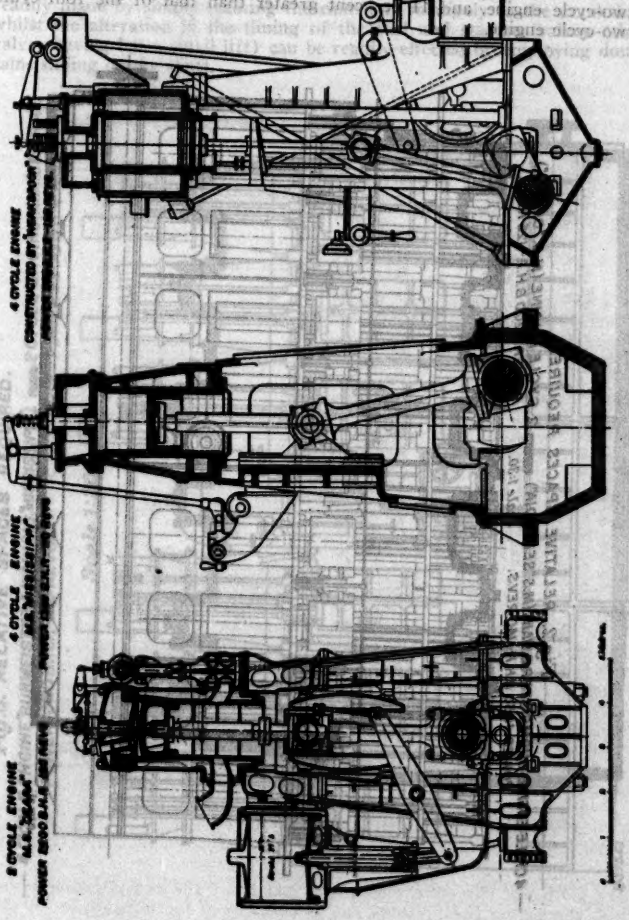



Fig. 1. RELATIVE IMAGES REQUIRED FOR VARIOUS TYPES.

and consequently of its fittings, such as supports, stern tube, etc. According to the rules of Lloyd's Register, the section of the shafting of a six-cylinder four-cycle engine (for the same power and the same number of revolutions) ought to be 45 per cent greater than that of the six-cylinder two-cycle engine, and 11 per cent greater than that of the four-cylinder two-cycle engine.

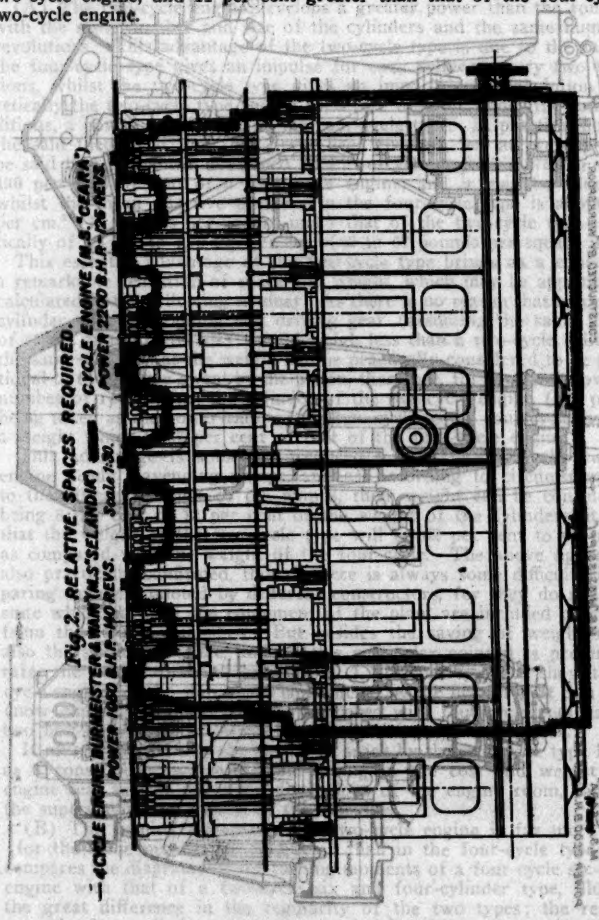

4 CYCLE ENGINE BURMEISTER & WAIN (M.S. "SELANDIA")
POWER 1060 B.H.P.—140 REVS.
2 CYCLE ENGINE (M.S. "CEARA")
POWER 2200 B.H.P.—126 REVS.
Scale 1:30.

Fig. 2. RELATIVE SPACES REQUIRED.

Furthermore, the reduced size of the flywheel in the two-cycle engine and the reduced space permits of placing the engine nearer the stern, not only saving in the length of the line of shafting but also increasing the space available on board for the cargo.

(C) The two-cycle engine offers greater facility in reversing as compared to the four-cycle type, which is due to the fact that in the former

the exhaust of the burnt gases takes place through ports in the cylinder wall, so that in order to reverse the running it is only necessary to alter the timing of the scavenging valves, of the fuel valve and of the starting valves. This alteration in the timing of the scavenging valves is very readily made by simply rotating the camshaft relatively to the crankshaft, whilst the alteration in the timing of the fuel and starting valves (these valves having but a small lift) can be readily effected by employing double cams sliding on the shaft.

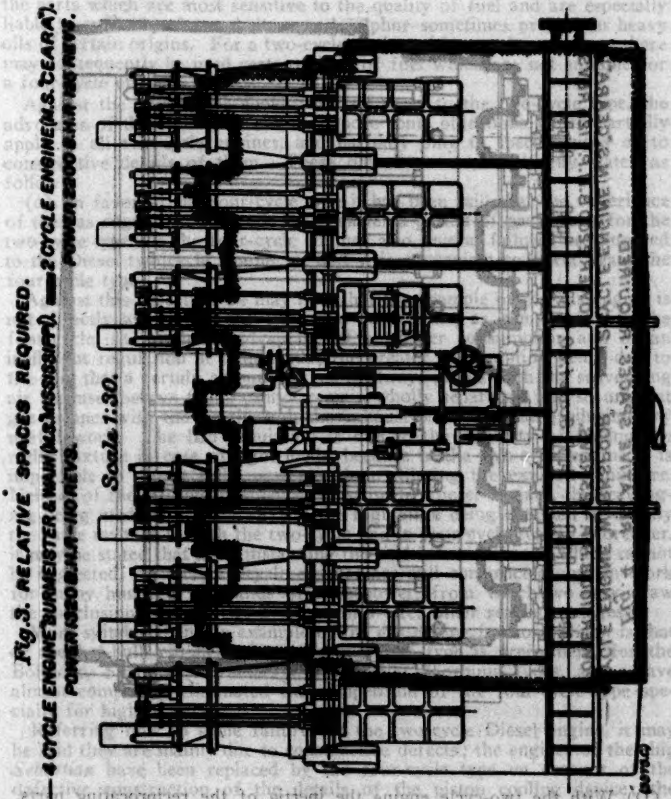


Fig.3. RELATIVE SPACES REQUIRED.
4 CYCLE ENGINE BURMEISTER & WAIN (M.S.MISSISSIPPI). 2 CYCLE ENGINE (M.S. CEARA).
POWER 1350 B.H.P.-110 REVS.
POWER 2200 B.H.P.-115 REVS.
Scale 1:30.

In the four-cycle type, on the contrary, besides the alteration in the timing of the fuel and starting valves, it is necessary separately to reverse the inlet and exhaust valves; and as the latter operation requires a different rotation of the camshaft, it is not possible to employ the simple device of the two-cycle type, but much more complicated mechanisms become necessary.

Referring, further, to the starting and reversing devices, it may be added that the necessity of being able to start the engine whatever be the position in which the crankshaft has stopped, that is to have at least one of the cylinders in the inlet phase of the starting air, does not permit of a reduction in the number of the cylinders to less than six in the four-cycle type, whilst the two-cycle can be constructed with but four and keep its perfect maneuverability.

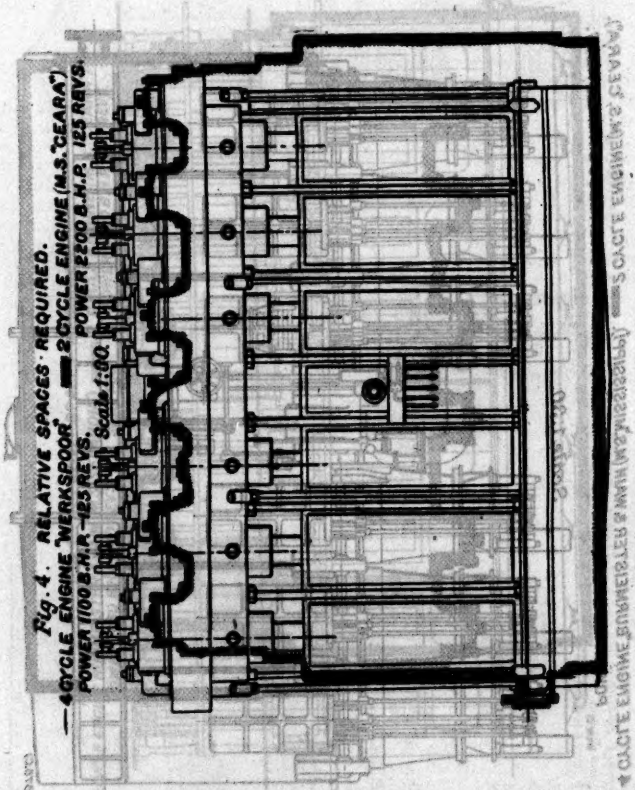


Fig. 4. RELATIVE SPACES REQUIRED.
—4 CYCLE ENGINE "WERKSPOOR" POWER 1100 B.H.P. —125 REVS.
—2 CYCLE ENGINE (M.S. "CEARA") POWER 2200 B.H.P. 125 REVS.
Scale 1:100.

(D) With the two-cycle engine the inertia of the reciprocating parts, such as connecting rods, pistons, &c., is balanced at top dead center by the pressure on the piston, which cannot be realized in the four-cycle for the exhaust and suction strokes. As a consequence, in the four-cycle engine the piston rods are subjected to alternative compressive and tensile stresses, so that the caps and bolts of the connecting-rod heads and of the main bearings must be necessarily constructed much more strongly than in the two-cycle type in order to avoid the possibility of their breaking and the great damage which this would cause.

(E) The two-cycle engine does not require any exhaust valve for the burnt gases, and in the engine provided with port scavenging there is no need of any valve subjected to the action of the burning gases; in the four-cycle type the exhaust valves are the source of well-known troubles and even in the case where their tightness and durability is increased by using more or less complicated cooling devices, the danger of their falling into the cylinder, with all its serious consequences, can never be fully eliminated.

It should be noted that the exhaust valves in the four-cycle engine are the parts which are most sensitive to the quality of fuel and are especially liable to suffer by the asphaltum and sulphur sometimes present in heavy oils of certain origins. For a two-cycle engine without exhaust valves there may consequently be used certain kinds of fuel which are not suitable for a four-cycle engine.

Against the advantages above referred to as to the two-cycle type, the advocates of the four-cycle engine oppose some objections which partially apply to all two-cycle engines, and partially only to special types or to constructive details of them. These objections may be briefly stated as follow:

(*a*) In favor of the four-cycle type it has been said that the experience of the gas engine has led back again (after a period of preference for the two-cycle type) to the four-cycle engine; also several failures are imputed to the Diesel two-cycle engine, so that it is convenient to select again the four-cycle type.

Against this objection we may note that the example of the gas engine is not directly applicable; the two-cycle gas engine, as compared with the four-cycle, shows the disadvantage of a greater consumption and of an inefficient regulation at light loads; the greater consumption being due to the fact that a certain amount of gas is always mixed with the scavenging air because the two fluids cannot remain wholly separated, and so unburnt gas escapes with the air through the exhaust ports without producing any useful work. The bad regulation is due to the difficulty of having the right mixture in case of light loads, because in the two-cycle engine it is impossible to regulate the power without diluting the explosive mixture. Neither of the said inconveniences exist in the Diesel engines, the scavenging being made with pure air and the regulation being obtained in exactly the same manner in both the two-cycle and in four-cycle types. Moreover, it may be stated that notwithstanding the said inconveniences, which cannot be neglected, the gas two-cycle engines are still constructed, and in work for many hundred thousands of horsepower, from which we may draw the conclusion that the two-cycle engines offer other real advantages.

More suitable than the example of the gas engine for comparison is that of the hot-bulb engines where the two-cycle type is preëminent, for the Bolinder, Skandia, Fairbanks-Morse, Petter, Torbinia types, a.s.o., have almost completely eliminated the competition of the four-cycle type specially for high powers.

Referring now to some failures of the two-cycle Diesel engine, it may be said they are mainly due to constructive defects; the engines of the ship *Sebastian* have been replaced by the four-cycle type on account of the defective construction of the details of the piston cooling device; the engines of the ships *Arum* and *Arabis* have shown defective lubricating systems; numerous inconveniences have been experienced in the engines with stepped pistons, and it would therefore be wrong to attribute these failures to the type of the engine in itself instead of to defects in the design.

Other failures ought to be attributed to the inexperience of constructors who, knowing but little of the two-cycle type, have risked building engines of great power and high speed. The two-cycle type of engine is not easy

to design; the scavenging of the cylinder is a very complicated problem, and it has not been possible to study it experimentally in all details although many trials have already been made, the most important of these being that of the Krupp firm, which has made a cinematograph record by using a glass cylinder. For scavenging it is necessary that the air current in the cylinder should fully and readily displace the combustion residues without mixing with them. The study and design of a convenient system of ports passages, as well as of suitable scavenging organs, in order to obtain a perfect scavenging, and at the same time the most reduced power consumption for driving air pumps, is a problem which can only be solved completely by many trials and long and costly experiments.

Should the cylinder and the scavenging ports not be well designed and the scavenging be imperfect, the working of the engine will be bad; instead of having the cylinder filled with pure air for the combustion, there will be therein a mixture of the air with the burnt gas not fully exhausted, so that the combustion will take place irregularly and be delayed, the fuel being therefore inefficiently utilized and at the exhaust such temperatures may be reached as may greatly reduce the durability of the cylinders and of the engine itself.

The two-cycle engines which are constructed at present by experienced manufacturers, and especially those with port scavenging, are by no means less reliable than the four-cycle type. For obvious reasons it is not possible to speak of engines installed on warships (the brilliant trip of a small Russian submarine from Spezia to Arkangel cannot be forgotten), but the success of the *Monte Penedo* and of the *Ceara*, which latter is the most powerful ship in service at present, are undeniable proofs that the two-cycle engine, if well constructed, can give the best results.

Considering the extensive use made in the German navy of combustion engines of both the four-cycle and the two-cycle types, it is noteworthy that the important company, Hamburger Werft A.G., recently founded by the A.E.G., and by the Hamburg-America Linie, for the construction of motor ships, under the management of the well-known Ballin, will give the preference to the two-cycle engine.

(*b*) The supporters of the four-cycle type allege that the two-cycle engines are far more complicated, not only on account of the scavenging pumps, the piping and the receivers relating thereto, but also on account of the greater complexity of the valve gear.

Against this assertion it may be objected that the air pumps which undoubtedly constitute an added organ, by no means interfere with the reliability of the working of the engine, as they are always working at very low pressures and temperatures like the low-pressure cylinders of steam engines; and constructively it is certainly more rational to employ a suitable air pump instead of using, for half the time, for displacing the air, enormous pistons which have been designed and fitted with rings for at least a hundred times higher pressure.

Referring now to the valve gear, the complexity pertains exclusively to that two-cycle type of engine having scavenging valves in the cylinder heads, whilst in the recent type with port scavenging, besides the fuel and the starting valve (like that of the four-cycle type), there is only the scavenging valve to control. This is light and easily displaced, as it is not subjected to the highest pressures and temperatures of the cycle, and it does not require to be perfectly tight. This valve can easily be replaced by a rotary valve. In the cylinder of the four-cycle engine, instead of one scavenging valve there are two at least to be controlled, and very often two inlet and two exhaust valves, which, being placed in the combustion chamber, require to be perfectly tight and need a precise and reliable operating gear in order to withstand the effort of the powerful closing springs.

(*c*) In favor of the four-cycle type it has been furthermore affirmed that its fuel consumption is far lower than that of the two-cycle engine. Now even if it must be admitted that this objection is correct in relation to the first two-cycle engines which were constructed, and is also applicable to some present motors of defective construction, it has, nevertheless, lost much of its importance when comparing the four-cycle engine with the best-known modern two-cycle engines.

For slow two-cycle engines the consumption may be reduced under 200 grammes (0.344 pound) per B.H.P. per hour. The consumption of 194 grammes per brake horsepower has been obtained since 1915 with the 2,200 brake horsepower two-cycle engines of the ship *Ceara* on a brake test, with the most careful observation and all auxiliary pumps (such as scavenging, compressing and water, or oil pumps) directly driven, working with heavy oil of a poor quality of the density of 0.90. For the high-speed engines we may cite figures comprised between 203 grammes (0.447 pounds) and 210 grammes (0.463 pounds) per brake horsepower as resulting from the official tests of many two-cycle engines of the power of 350 brake horsepower at 480 revolutions and of 1,350 brake horsepower at 350 revolutions.

The following are the figures for consumption of fuel obtained in recent official tests on an Ansaldo San Giorgio engine of 1,300 H.P., 350 revolutions, furnished to an allied Government.

| | Horse-power. | Revolutions. | Gr. |
|---|---|---|---|
| At 4/5 of power | 1,110 | 292 | 194 |
| At full power | 1,339 | 352 | 203.1 |
| With 10 p.c. over power | 1,425 | 356 | 202.9 |

The consumption of lubricating oil was about 6 grammes (0.013 pounds) per horsepower per hour, although the lubrication was very abundant inasmuch as the motor was new.

The consumptions ascertained for the four-cycle engines are not much different; the data which have been published as to the slow marine four-cycle engines shew 182 grammes to 188 grammes (0.401 pounds to 0.414 pounds) per brake horsepower, often excluding the consumption of the auxiliary pumps which were separately driven. With the high-speed type the competition of 1913 of the German Admiralty for electric sets developing 300 kw. at 400 revlutions, in which the most important manufacturers specializing in the construction of the Diesel engines took part (for instance, M.A.N., Krupp, Koerting) has led to figures, which were officially published in "Der Oelmotor," 1913, and according to which the consumption in the four-cycle engines was from 197 grammes to 203 grammes (0.434 pound to 0.447 pound) per B.H.P. per hour.

It is true that some excessively low figures have been singly reported for the consumption of four-cycle engines, but they can be safely overlooked upon consideration of the circumstances of the test or of the uncommonly high consumption of the lubricating oil, which was obviously partially burnt as fuel, so that the above-stated results can be quoted as corresponding to the best up-to-date constructions. Though they still show a slight advantage for the four-cycle engine, this is not greater than 3 per cent or 5 per cent, and if we consider the other elements required for calculating the real working expenses, this difference is not of great importance. It must, indeed, be noted that the installation of two-cycle instead of four-cycle engines for a given type of ship, results in a saving in weight and space, and therefore a reduction of displacement and the possibility of increasing the run of the stern (this leading to a reduction in the power

for propelling the ship at a certain speed). This advantage amply compensates for the slightly greater fuel consumption, specially in high-speed ships where the weight and the space taken by the propelling plant have the greatest influence. Furthermore, it may be added that, even if the question of the weight and space should be regarded as a secondary one, still the two-cycle engine shows the advantage that the particulars being the same, it can develop the same power as the four-cycle one at a much lower of speed of revolution, with the consequence of a reduction in the consumption and of a much better efficiency of the propeller. It is also very likely that as the consumption in the two-cycle engines has decreased, as a consequence of rational and systematic experiments, in a few years, from 350 grammes or 260 grammes per brake horsepower, to the present values, it will still improve until it reaches and even surpasses the low consumption of the four-cycle type. Theoretically there is no reason why this should not happen, for the thermal efficiency is the same in both types, and the power required by the two-cycle engine cannot be greater than the power expended in driving the main pistons of the four-cycle engine to work half the time as pumps themselves.

Finally, besides the fuel consumption, that of the lubricating oil, which is much more expensive, ought to be considered. It is obvious that the two-cycle engine should require a less quantity of oil than the four-cycle, the load on the piston of the four-cycle engine being 50 per cent greater (with the same number of cylinders and the same ratio between diameter and stroke) than that of the two-cycle, the pressure exerted on the bearings, and on the guides being proportionally increased, so that the surfaces to be lubricated are accordingly larger. In practice, however, as the two-cycle engine may be constructed with fewer cylinders the saving in the lubricating oil is still more evident. At present the figure of 3 grammes to 4 grammes (0.00614 pounds to 0.008818 pound) per brake horsepower as the total amount of oil consumption is usually reached in high-speed engines (480 revolutions).

(*d*) As another advantage of the four-cycle type it is affirmed that the cylinder wall never reaches such high temperatures as in the two-cycle type, so that the latter are subjected to higher internal strains and thus to the danger of cracks. Now, whilst it is true that the ratio between the quantity of the fuel burnt in the four-cycle type and the surface of the combustion chamber is hardly superior to one-half of the same ratio in the two-cycle engine, other important circumstances have been overlooked which have certainly a great influence on the mean temperatures.

The action of the hot gases on the cylinder walls lasts certainly a shorter time in the two-cycle than in the four-cycle type. Whilst in the latter the cylinder walls undergo the action of the hot gases during the whole expansion and exhaust strokes, that is, practically for more than half the time, in the two-cycle engine the action of the hot gases lasts only for little more than two-thirds of the working stroke.

In two-cycle engines in which the exhaust occurs through ports, the latter open much more rapidly than the exhaust valves of the four-cycle engines, and consequently there is a much more rapid diminution in the temperature due to expansion.

Whilst the exhaust temperature in four-cycle engines is seldom below 350 degrees C. and in the high-speed engines it easily reached 450 degrees or 500 degrees C., in two-cycle engines, if well constructed, this temperature usually remains under 250 degrees C., and sometimes it only reaches 200 degrees or 210 degrees C.

Taking account of this fact a diagram has been drawn of the relative temperatures of the high-speed four and two-cycle engines (Fig. 6), which has no absolute value but merely a relative one. It is based for both types on the same hypothesis, that is, volumetric efficiency 80 per cent,

referring to air at the atmospheric pressure and at 15 degrees C., adiabatic compression from the initial temperature of 100 degrees C.; combustion at a constant pressure; adiabatic expansion, mean effective pressure 5 kg. per cm.$^2$ for the four-cycle and 4.375 kg. per cm.$^2$ for the two-cycle engine (corresponding to the ratio of 1.75 between the volumes of piston displacement); fuel consumption 198 grammes per brake horsepower for the four-cycle and 208 grammes for the two-cycle type; exhaust temperature of 450 degrees for the four-cycle engine (being the average temperature as ascertained in the high-speed engines presented at the competition of the German Admiralty above referred to). For the exhaust stroke in the four-cycle engine the curve of the temperatures is such as practically ascertained according to Dr. E. B. Wolff's experiments (see "Oelmotor," 1915), where for the suction stroke a gradual mixing of the residual exhaust gases in the compression space with the atmospheric air has been assumed.

*Fig. 5.* DIAGRAM COMPARING THE TURNING MOMENTS OF TWO-CYCLE AND FOUR-CYCLE ENGINES.

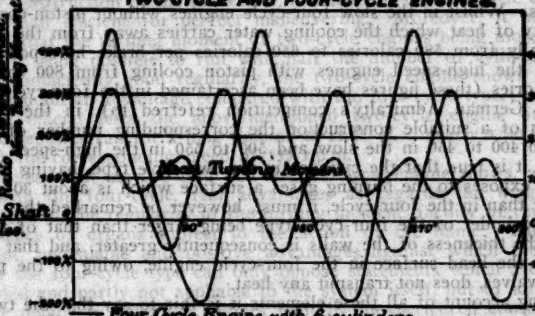
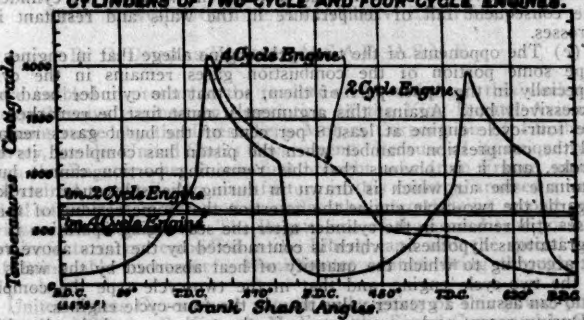



*Fig. 6.* DIAGRAM COMPARING THE TEMPERATURES IN THE CYLINDERS OF TWO-CYCLE AND FOUR-CYCLE ENGINES.



Not one of the hypotheses above referred to is in favor of the two-cycle engine; the hypothesis of the same initial compression temperature in both types is unfavorable for the two-cycle type, as all experiments which have been made with gas engines confirm that in the two-cycle engines a much higher compression ratio can be employed than in the four-cycle, without the danger of premature ignition, and that the mixture at the beginning of the compression is therefore cooler in the two-cycle type. By measuring the diagrams by a planimeter, however, the conclusion was reached that the mean temperature of the two cycles is practically the same.

The above result is confirmed by measuring the quantities of heat absorbed by the circulating water in the two-cycle and in the four-cycle engines. Whilst in the slow four-cycle engines without piston-cooling the quantity of heat which the cooling water carries away from the cylinders is usually from 580 calories to 650 calories per brake horsepower hour, and in the high-speed engines with piston cooling from 800 calories to 850 calories (these figures have been ascertained in the four-cycle engines of the German Admiralty's competition referred to), in the two-cycle engines of a suitable construction the corresponding number of calories is from 400 to 450 in the slow and 500 to 550 in the high-speed engines. Whilst it is true that the cylinder in the two-cycle type, having the same power, exposes to the burning gases a surface which is about 30 per cent smaller than in the four-cycle, it must, however be remarked that the size of the cylinder of the four-cycle type being larger than that of the two-cycle, the thickness of the walls is consequently greater, and that a great part of the head surface in the four-cycle engine, owing to the presence of the valves, does not transmit any heat.

Taking account of all these elements it is fair to say that the two-cycle engine, from the standpoint of temperature, is in better condition than the four-cycle. The two-cycle engine, in which the inner walls of the cylinder, after the very short action of the flame, are immediately cooled by the scavenging air current (which is supplied in such quantity as to allow, besides the filling up of the cylinder, the escape of the warmest portion which entered at first) is thermally superior to the four-cycle engine in which all heat must be abstracted through the walls of the cylinders with the consequent fall of temperature in the walls and resultant internal stresses.

(*e*) The opponents of the two-cycle engine allege that in engines of this type some portion of the combustion gases remains in the cylinders, especially in the upper part of them, so that the cylinder head becomes excessively hot. Against this argument it must first be remarked that in the four-cycle engine at least 8 per cent of the burnt gases remain and fill the compression chamber when the piston has completed its exhaust stroke, and it is obvious that this remaining portion cannot but contaminate the air which is drawn in during the subsequent stroke. As regards the two-cycle engine the assertion that some residue of the burnt gases still remains in the cylinder after the scavenging operation is merely a gratuitous hypothesis, which is contradicted by the facts above referred to, according to which the quantity of heat absorbed by the walls is less in the two-cycle engine, and that in the two-cycle type the compression ratio can assume a greater value than in the four-cycle engines.

During recent and accurate tests in the test room of the Ansaldo San Giorgio works on a high-speed two-cycle engine, it has been determined that the quantity of carbon dioxide contained in gas enclosed in the cylinder during the return stroke does not exceed 0.3 per cent, and that the contents of oxygen is hardly less than the content in the pure atmospheric air, *i.e.*, 20.5 per cent instead of 20.9 per cent.

(*f*) Against the two-cycle engine it has been said that the four-cycle type can run with greater regularity than the two-cycle when working at

low speed of revolutions, owing to the fact that in the two-cycle engine the compression at low speed falls rapidly with the diminishing of the scavenging air pressure. It must, however, be noted that this observation is correct merely when it refers to two-cycle engines of bad design, in which, owing to inefficient construction, the scavenging air pressure rises, at the normal speed, to excessively high value, whilst in two-cycle engines, which have been carefully designed, even at full speed the pressure of the scavenging air remains within very small limits. By the speed reduction the pressure is also somewhat reduced, but not so as to cause failure of the ignition especially when the engine is hot. Practically, in both the two-cycle and in the four-cycle types the lowest limit of speed is dependent upon the construction of the pulverizer, and this for the two-cycle engines is more than efficient for perfect maneuvering. Moreover, it must be remarked that the turning moment of two-cycle engines being more regular, and it being possible to run with half the number of cylinders and to obtain sufficiently good regularity, the two-cycle engine shows in this particular point an advantage compared with the four-cycle type.

During official tests the above-mentioned two-cycle engine of 1,300 brake horsepower ran for a long time at 45 H.P. (*i.e.*, nearly 1/30 power) and with the corresponding speed of 115 r.p.m.

In conclusion, it may be said that the two-cycle engine shows, as compared with the four-cycle, real advantages as regards the weight, the space required, the regularity of the turning moment, facility of reversing, and absence of organs subject to the action of the flame. These are important, undeniable and positive advantages against which the advocates of the four-cycle type can only oppose statements which are partly unfounded and partly not appliable to the system itself, but to some constructive details in defective engines built by inexperienced constructors. Judging by the tests and practical results of the two-cycle engines of good construction the writer holds that this type should become standard for the Diesel marine engines, as it is already for the hot-bulb engines.—"Engineering."

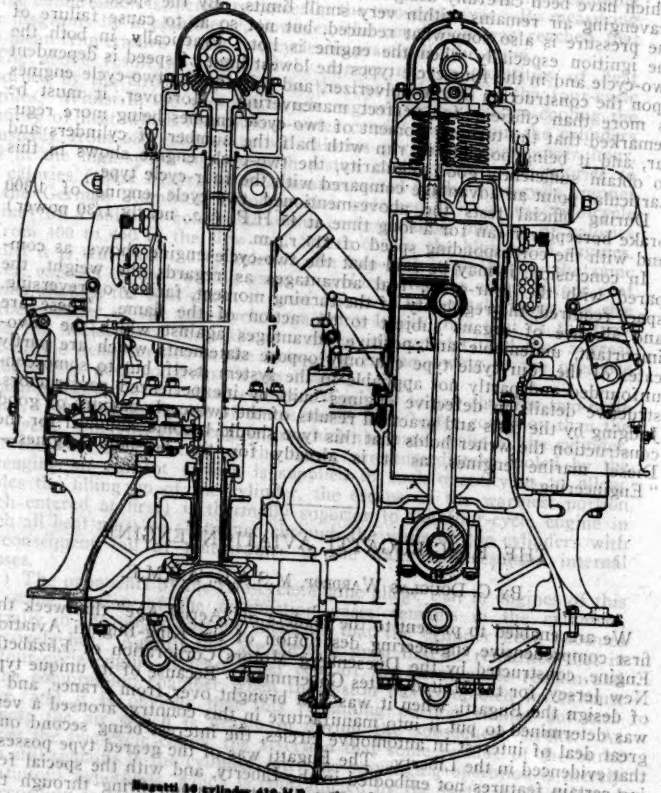
---

## THE KING-BUGATTI AVIATION ENGINE.

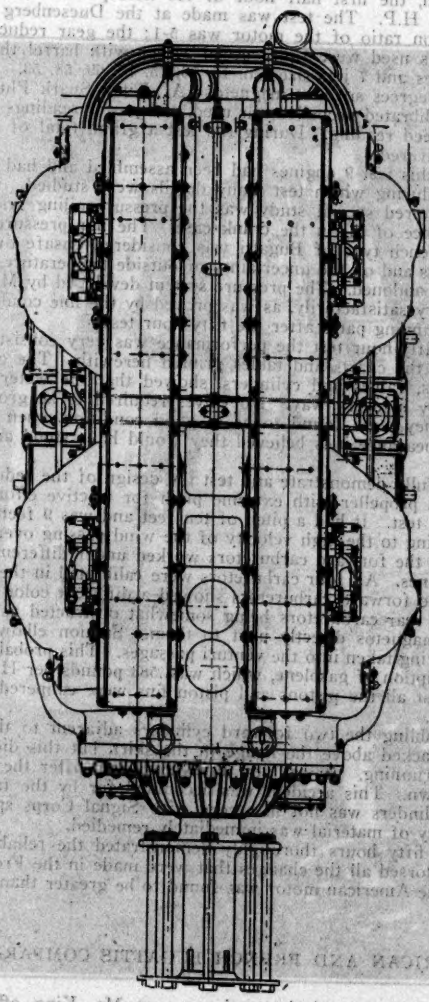
By G. Douglas Wardrop, M.S.A.E., A.S.M.E.

We are enabled to present to the readers of "Aerial Age" this week the first comprehensive engineering description of the King-Bugatti Aviation Engine, constructed by the Duesenberg Motors Corporation of Elizabeth, New Jersey, for the United States Government. Because of its unique type of design the Bugatti, when it was first brought over from France, and it was determined to put it into manufacture in this country, aroused a very great deal of interest in automotive circles, the interest being second only that evidenced in the Liberty. The Bugatti was of the geared type possessing certain features not embodied in the Liberty, and with the special feature of being able to mount a 37-millimeter cannon firing through the propeller shaft.

The United States Government put the redesigning of this power plant in the hands of Charles B. King, A. M. E., of the Signal Corps and he was made responsible for all changes. With a large force of draftsmen the work of adjusting the Bugatti type of design to American manufacturing methods was undertaken and upon the completion of the redesigned motor Mr. King transferred to the Aircraft Production Board a report which considered carefully all the details which are discussed hereafter and in which only minor changes have been made, since the report was handed in, which emphasizes the good judgment used when the motor was being redesigned.



Bugatti 16 cylinder 410 H.P. engine—Sectional view.



King-Bugatti 410 HP. Engine—top view

SIXTEEN CYLINDER KING—BUGATTI AVIATION ENGINE.

The test referred to was one of fifty hours' endurance, ten periods of five hours each, the first half hour at 410 horsepower and 4 hours 30 minutes at 380 H.P. The test was made at the Duesenberg test room. The compression ratio of the motor was 5-1; the gear reduction 28:42. The carburetors used were four special Miller with barrel throttle, with choke 1⅞ inches and 7 jets size No. 76, 76, 75, 72, 69, 58, 53. Four Dixie Model 800-38 degrees advance magnetos; AC Titan Spark Plugs, two per cylinder. A calibrated propeller was used, and power readings used corresponding to speed reading. During the test a grand total of 19,284 H.P. hours were delivered.

Previous to this test 9 engines had been assembled and had run a total of 110 hours, during which test many details were studied. One of the points that required special study was the pressure oiling system due to cross interference of oil in the crank case. The non-pressure system as used in the French type of Bugatti was considered unsafe owing to the many long leads and on the uncertainty of outside temperature, and it was consequently abandoned. The pressure system developed by Mr. King has worked out very satisfactorily, as was proved by the fine condition of the bearings and running parts after the fifty-hour test.

During the fifty-hour test the performance was very consistent, as will be noted from the charts and tables printed herewith. The condition of the spark plugs, valves and cylinders, showed that the water circulation was satisfactory in every way. No valves required to be ground during the test and they were found to be in good condition when dissembled. From their appearance it is believed they would have stood an additional fifty hours.

In order to fully demonstrate and test the design of the reduction gears and bearings a propeller with extreme pitch for tractive effort was used throughout the test. It had a pitch of ten feet and was 9 feet 4 inches in diameter. Owing to the high velocity of the wind passing over the engine during the test the forward carburetors worked under different conditions than the rear ones. All four carburetors were calibrated in the jets before the test, but the forward carburetors showed a different color of flame at the stacks, the rear carburetors being somewhat obstructed as to air currents by the magnetos directly next to them. Suction elbows were not used, the air being taken into the venturi passages. This probably accounts for the consumption of gasolene, which was .583 pounds per H.P. hour.

After the test all the pistons and piston fins were calipered and found round and true.

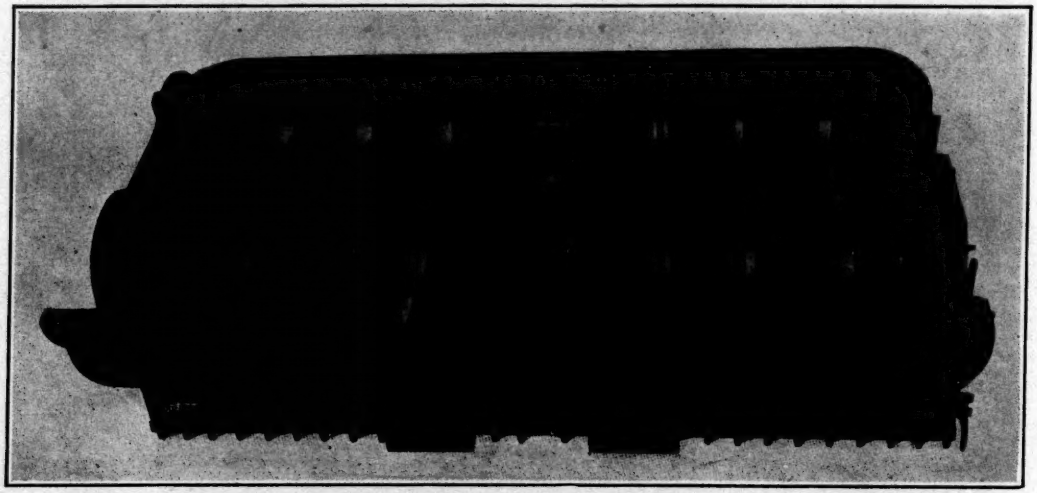
Upon dissembling the two forward cylinders adjacent to the propeller were found cracked above the flange in the skirt, but this did not interfere with the running. It was not discovered until after the engine had been taken down. This accident was accounted for by the fact that the iron of the cylinders was not in accord with Signal Corps specifications. The discrepancy of material was immediately remedied.

The test of fifty hours thoroughly demonstrated the reliability of the engine and endorsed all the changes that were made in the French design. The H.P. in the American motor was found to be greater than that of the French.

## AMERICAN AND FRENCH BUGATTIS COMPARED.

In the course of a very interesting report Mr. King offers definite data concerning the specific changes made in the Bugatti design and the reason therefor. Owing to the fact that the French engine which was sent over to this country had had a limited test in Paris of 37 hours and had not been in flight, all of the points in the design were very carefully

SIXTEEN CYLINDER KING—BUGATTI AVIATION ENGINE.



CRANK CASE KING—BUGATTI ENGINE.

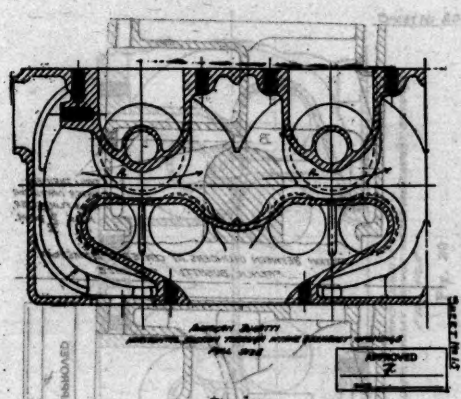


Fig. 15

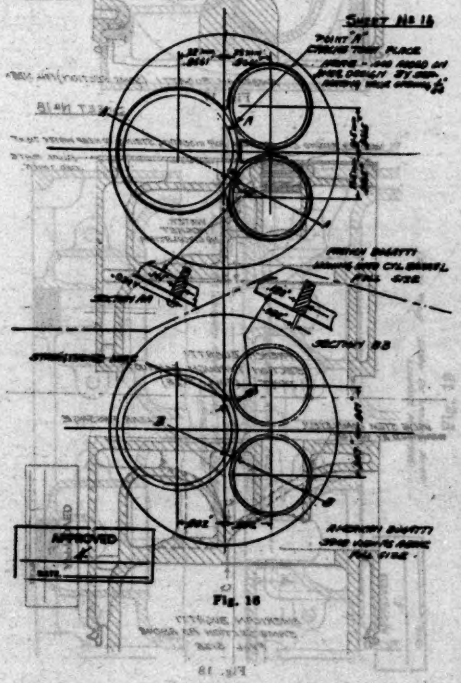


Fig. 16

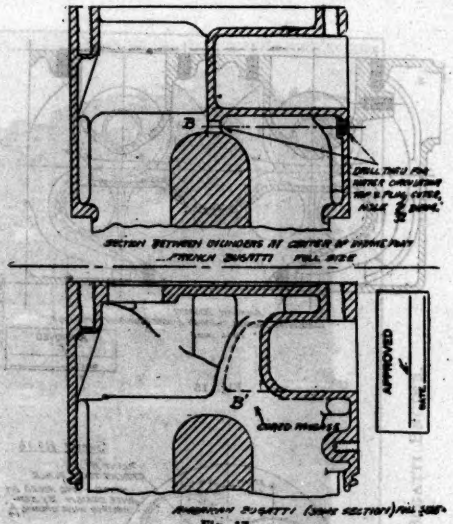



Fig. 17

SHEET No 18

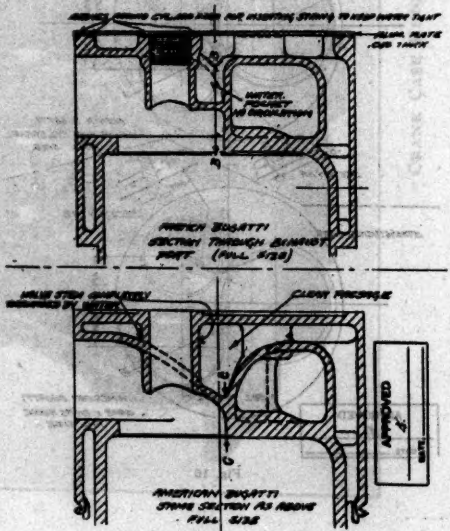



Fig. 18

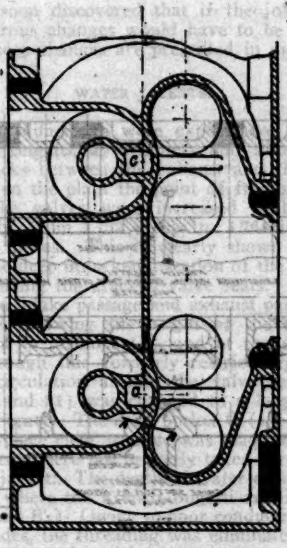


Fig. 20

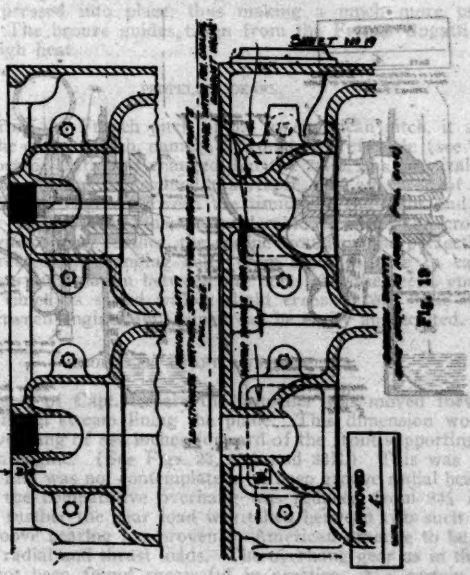


Fig. 19

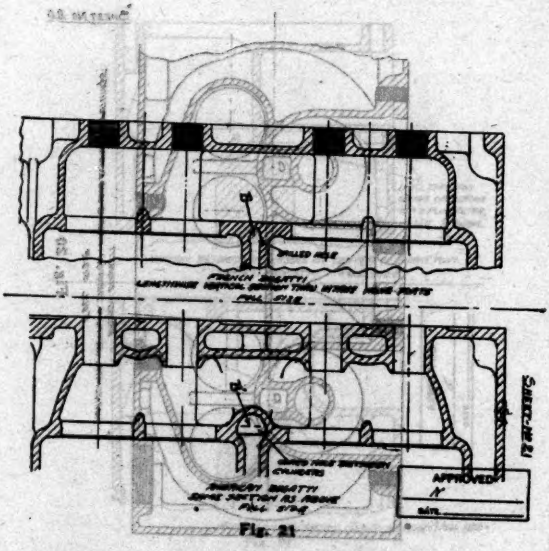



Fig. 21

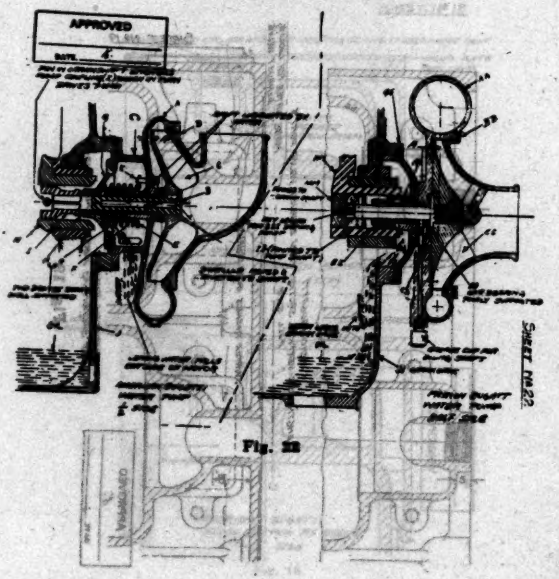



Fig. 22

considered. It was soon discovered that if the job was to be made a production one numerous changes would have to be made. Some of the more important of these changes are presented in diagram.

WATER JACKETS.

It was evident that difficulties were experienced in cooling the valve seats of the French Bugatti, as cylinders and sample sections of used cylinders showed cracks between the exhaust and the inlets at Point A. See Fig. 16 showing on the plant the point of fracture. In the American design, centers of the valves were increased in both directions from center of line, namely, from 1.732 inches to 1.764 inches and also from 1.693 inches to 1.694 inches. This is clearly shown on Fig. 16. Section AA, Fig. 20, shows the deep unjacketed section of the cylinder in which the strain was set up, causing the cracked valve seats. To obviate this difficulty, the shape of the intake passage and exhaust passages were improved and the distance BV, showing the height of the unjacketed wall (see Fig. 18) was reduced to distance CC, and at the same time circulation water was swept through this formerly restricted passage. To further improve the water circulation around the valve seats the small drilled hole B (see Fig. 17 and 21) was changed to a larger cored passage B', shown on this same Figure. The dead pocket C (see Fig. 20) was entirely eliminated and in its place clear passage was made as at D (see Fig. 15). The exhaust-valve stems were not properly taken care of as to heat transference to the water jacket. The depth of water around valve-stem guides is shown on Fig. 19, where the French Bugatti and American Bugatti are contrasted. (See B and B'.) Owing to poor conductivity through threaded portions of valve guides, the threading was eliminated and cast iron valve guides were pressed into place, thus making a much more uniformly cooled stem. The bronze guides taken from the French Bugatti showed evidence of high heat.

PROPELLER GEARS.

In transferring the French pitch of 5.6 to American pitch, it was considered that the coarser pitch, namely, 5, would be preferable (see Fig. 32), and the number of teeth was changed, but the ratio was practically maintained. It was further determined that there was an error of 4°10' in the French Bugatti engine between the timing of the right and the left crankshafts. As this would have considerable effect in the cross firing from the magnetos and, further, the engine would not be perfectly timed, this was corrected by changing belts from eight to nine in each gear. This with a certain relation between the holes and the teeth, enables the engine to be timed as stated with a slight error of 20'. The difference in running between engines thus timed can be easily appreciated.

PROPELLER SHAFT AND BEARINGS.

At the request of Capt. Lepere the propeller was moved forward 3¼ inches to assist in stream lining the plane. This dimension would have brought an overhang of 8¾ inches forward of the front supporting bearing in the French engine. (See Figs. 23, 23A and 23B.) This was far from good practice and was not contemplated. A deep groove radial bearing was selected and the comparative overhang was reduced from 8¾ inches to 3 inches and further the gear load was taken between two such bearings. This deep groove bearing has proven in American practice to be good, as it takes both radial and thrust loads. The overhang gear as in the French engine has not been found successful in practice, as a certain wedging

action takes place from the gear load and forces the true pitch contact line to assume a diagonal line. The ball-bearing thrust as used in the French engine permits the balls to be thrown outward by centrifugal force and causes an improper contact on ball surfaces; this can be further aggravated by the wedging action of the gears. In the parts removed from the broken up French engine the thrust rings and balls were burned to a deep blue. The assembly of the propeller-shaft bearings is a difficult "in place" assembly in the French engine. In the American design the bearings are entirely an "outside" assembly. The sliding door in front cover of the French engine has been eliminated.

WATER PUMP.

The French Bugatti pump as mounted on the engine permits water leakage to enter the sump and mix with the lubricating oil. This will lead to difficulties in the operation of the engine. (See Fig. 22.) In the American model the pump was moved back, a better support added (see distances C and C') and an opening was arranged so that the leakage could pass outside of engine sump. Two bearings were provided on pump shaft. The rotor is riveted to shaft and driven with a Woodruff key. Lubrication is provided with water which passes through center of shaft. It is found in practice in draining the system in cold weather that where the rotor is brought clear to the housing wall (see Dimension A) that water is held by capillary attraction in this narrow space and freezing results at this point and when the engine is started the pump shaft is sheared off. This space has been opened up in the American model (see Dimension A') and the water is thus permitted to freely pass out. An elbow has been added to pump cover which can be assembled pointing in any desired direction. It is believed in this redesign that the troubles experienced during the official test have been eliminated.

CONNECTING ROD ENDS AND LINERS.

The French Bugatti engine had no liners in connecting rod ends (see Fig. 24) and depended on a thickness of babbitt of .014 inch directly on the steel shell. This being an extremely thin layer, a considerable risk was offered in a 16-cylinder engine. In case a rod was burned out steel would come in contact with steel, resulting in a seizure of rod on crank pin, a broken rod would follow, meaning the ultimate wreck of engine. In the American engine, bronze liners were put into rod ends and .047 inch of babbitt was used. In case of a burned-out rod, steel will come in contact with the bronze and the liner therefore will not seize on the crank pin. This will enable the flier to still operate his engine and make a landing. Owing to the oil pressure in the American engine, the long grooves in bearings are eliminated and in their place the short grooves are used. The replacing of liners in rods is comparatively easy and does not demand a quantity of spare connecting rods for repairs.

ROCKER ARMS.

These are shown in detail on Fig. 24A. Simplicity of machining was studied and from 14 machining operations on the French Bugatti this has been reduced to 4 on the American model. Each engine has 48 rocker arms; therefore the French engine has 672 machine operations as against 198 on the American model.

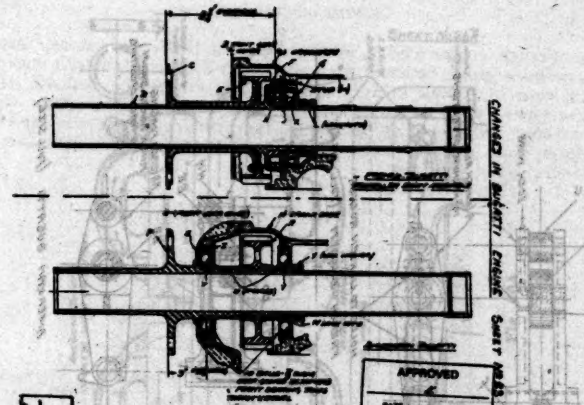


Fig. 23

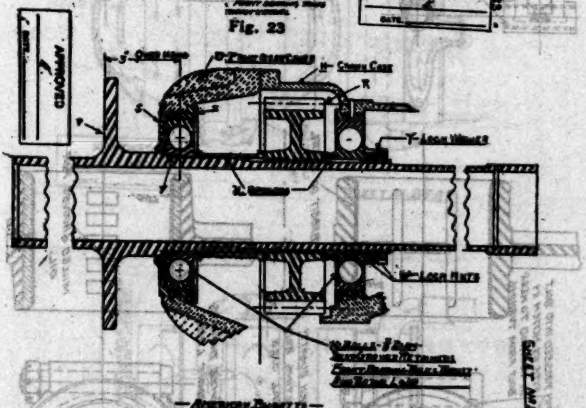


Fig. 23A

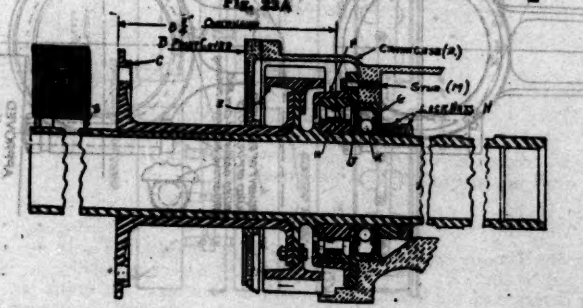


FIG. 23B

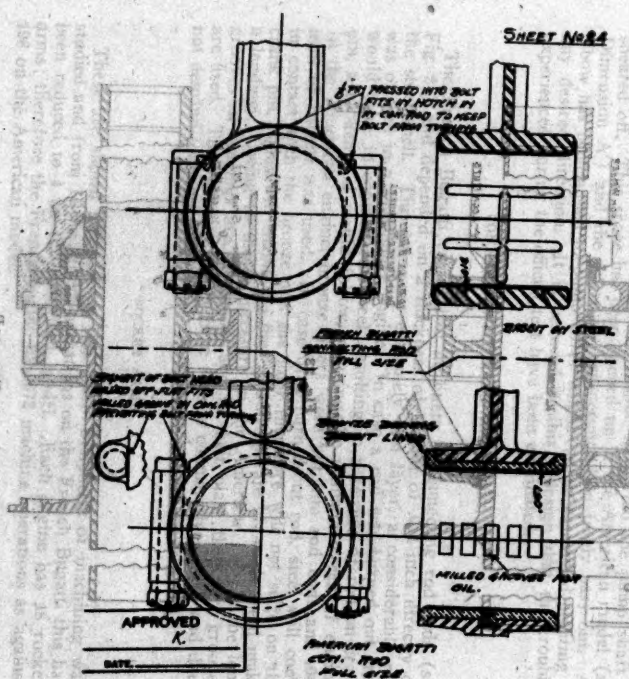



Fig. 24

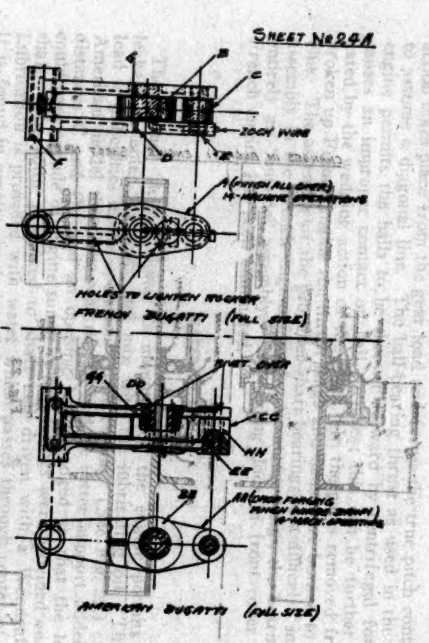



Fig. 24A

NOTES.

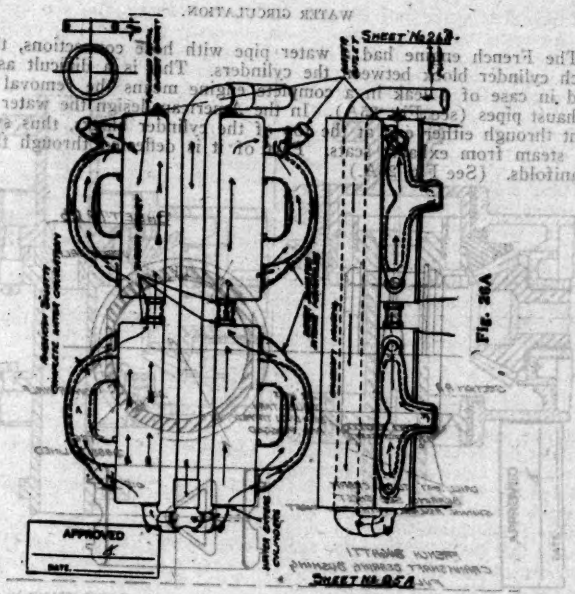


Fig. 28A

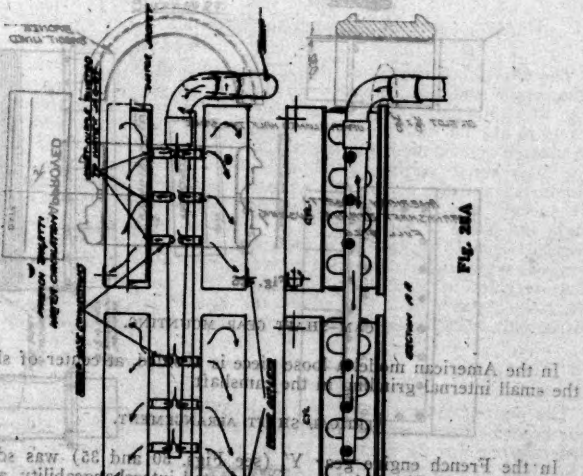


Fig. 28A

### WATER CIRCULATION.

The French engine had a water pipe with hose connections, three to each cylinder block between the cylinders. This is a difficult assembly, and in case of a leak in a complete engine means the removal of the exhaust pipes (see Fig. 25A). In the American design the water can be sent through either end at the top of the cylinder blocks, thus sweeping all steam from exhaust seats. Part of it is deflected through the inlet manifolds. (See Fig. 26A.)

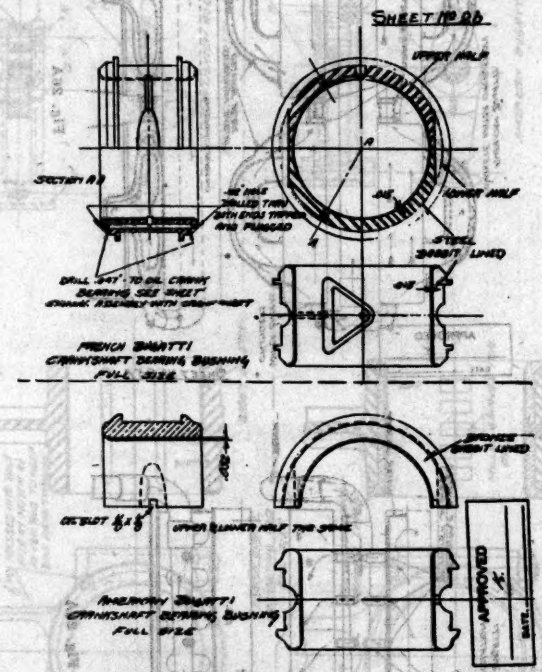


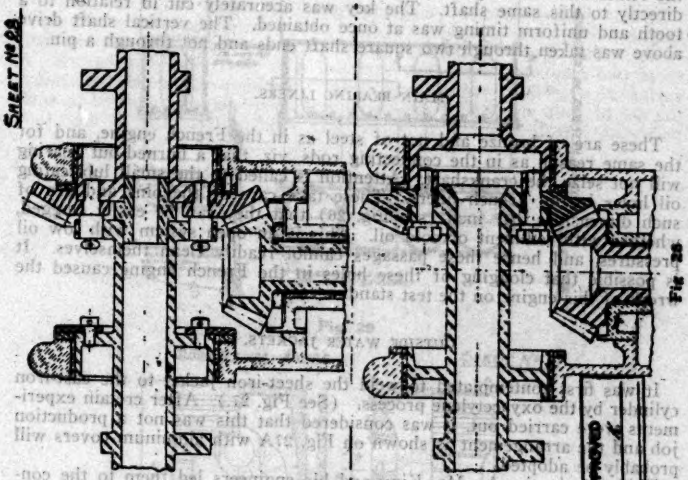

Fig. 26

### CAM-SHAFT GEAR MOUNTING.

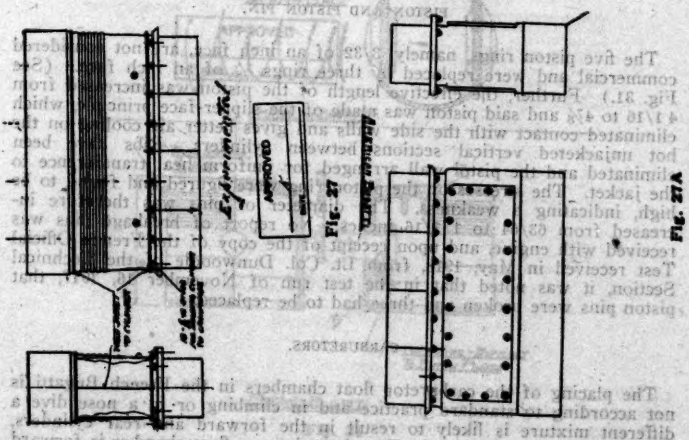
In the American model a loose piece is avoided, at center of shaft; also the small internal grinding in the camshaft.

### VERTICAL SHAFT ARRANGEMENT.

In the French engine gear Y' (see Figs. 30 and 35) was screwed on the end of shaft Z'. This at once prevented interchangeability, as it in no way determined the location of a tooth in relation to the crank pin. This in each individual French engine was to be corrected by proper timing




CHANGES IN BUGATTI ENGINE SHEET No 27



and then drilling the hole for pin N in vertical shaft. This was avoided in the American model by extending key to crank cheek ZZ fitting the gear directly to this same shaft. The key was accurately cut in relation to a tooth and uniform timing was at once obtained. The vertical shaft drive above was taken through two square shaft ends and not through a pin.

MAIN BEARING LINERS.

These are of bronze and not of steel as in the French engine, and for the same reason as in the connecting rods, viz, that a burned-out bearing will not seize the crankshaft. Attention is called to the small lubricating oil holes in the French liners. These take care of the pins and are of such dimensions .047 inch (see Fig. 26) that they can be easily clogged, whether with sediment or cold oil. This is an open system with low oil pressures, and hence these passages cannot readily clean themselves. It is possible that clogging of these holes in the French engine caused the wreck of this engine on the test stand.

OUTSIDE WATER JACKETS.

It was first contemplated to weld the sheet-iron jacket to the cast-iron cylinder by the oxyacetylene process. (See Fig. 27.) After certain experiments were carried out, it was considered that this was not a production job and the arrangement as shown on Fig. 27A with aluminum covers will probably be adopted.

This examination by Mr. King and his engineers led them to the conclusion that the Bugatti engine as delivered by the French to the American government was not a commercial engine, as it was necessary to redesign the entire motor and to bring it in line with American practice.

PISTON AND PISTON PIN.

The five piston rings, namely, 3/32 of an inch face, are not considered commercial and were replaced by three rings ⅛ of an inch face. (See Fig. 31.) Further, the effective length of the piston was increased from 4 1/16 to 4⅛ and said piston was made of the slipper-face principle, which eliminated contact with the side walls and gives better air cooling on the hot unjacketed vertical sections between cylinders. Ribs have been eliminated and the pistol wall arranged for uniform heat transference to the jacket. The stresses on the piston ring were figured and found to be high, indicating a weakness. The diameter of pins was therefore increased from 63/64 to 1 1/16 inches. No report of breakage pins was received with engine, and upon receipt of the copy of the French Official Test received in May, 1918, from Lt. Col. Dunwoodie of the Technical Section, it was noted that in the test run of November 16, 1917, that piston pins were broken and three had to be replaced.

CARBURETORS.

The placing of the carburetor float chambers in the French Bugatti is not according to standard practice and in climbing or in a nose dive a different mixture is likely to result in the forward and rear cylinders. (See Figs. 33 and 34.) In the French engine one float chamber is forward of the jet and one is in the rear. In the American engine they are both brought to the rear. The American engine therefore gets a uniform mixture in all cylinders.

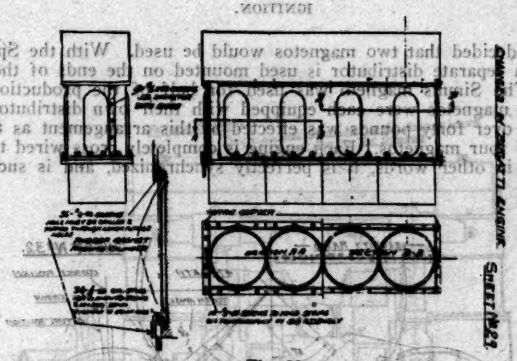


Fig. 29

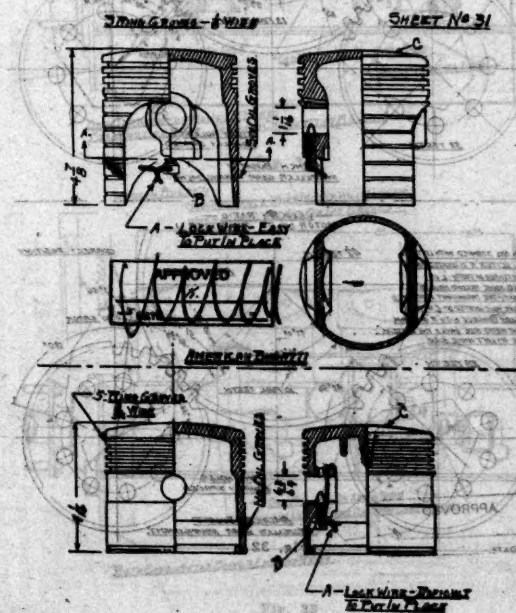


Fig. 31

IGNITION.

It was decided that two magnetos would be used. With the Splitdorf magneto a separate distributor is used mounted on the ends of the camshafts. The Simms magneto was used for half of the production, and these two magnetos were each equipped with their own distributors. A saving of over forty pounds was effected by this arrangement as against the use of four magnetos. Each engine is completely cross wired to each magneto; in other words, it is perfectly synchronized, and is such that

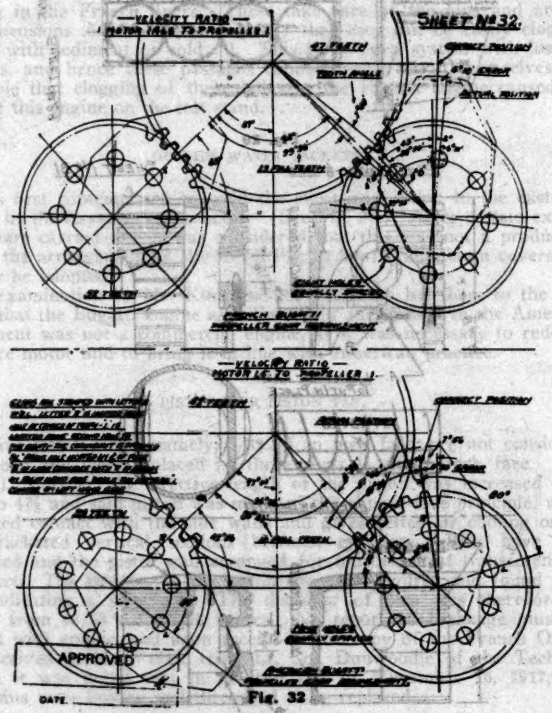



Fig. 32

each magneto taken separately will operate engine, and owing to this perfect synchronization the two magneto arrangement runs with less vibration than the four-magneto scheme. With the two magnetos and the distributor a starting magneto is supplied. This consists of a hand-operated independent magneto in the fuselage, which delivers a shower of sparks on the proper cylinder as selected by the distributor. Ease of starting is facilitated in this manner.

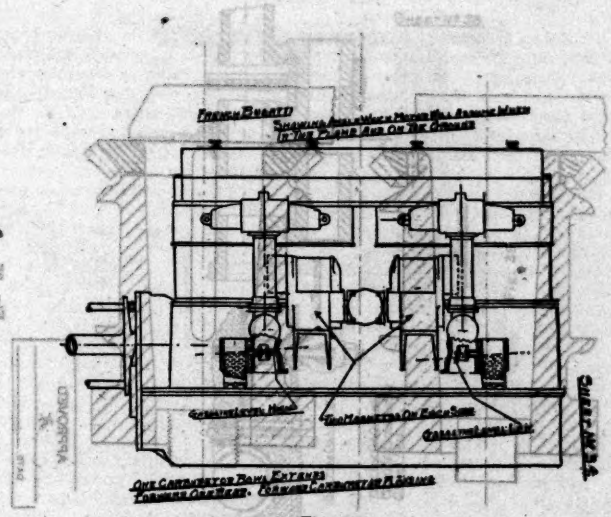


Fig. 34

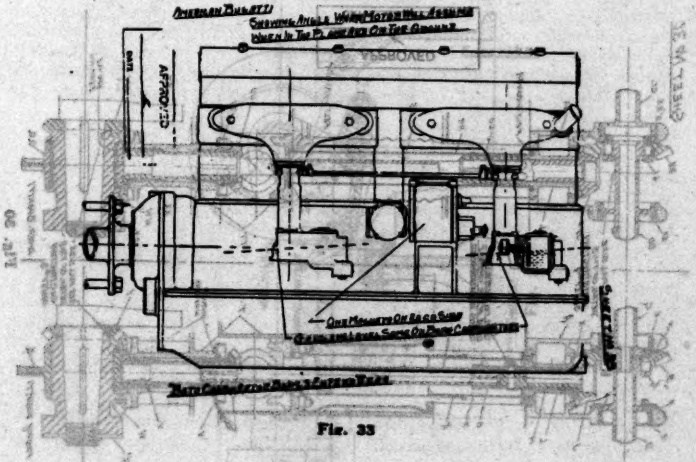


Fig. 33

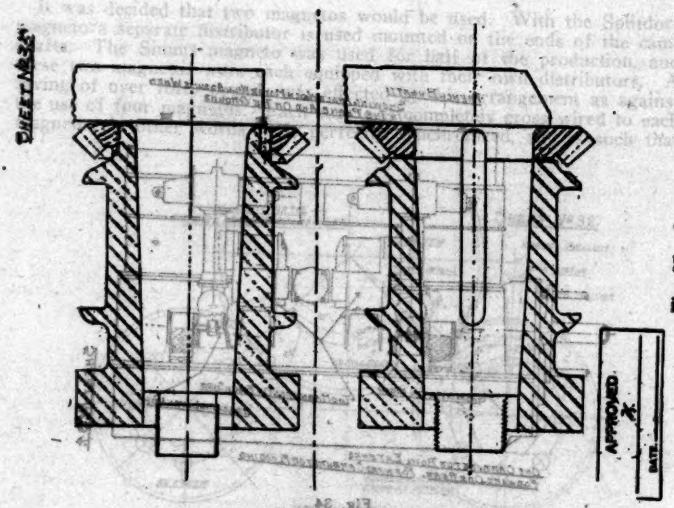



Fig. 35.

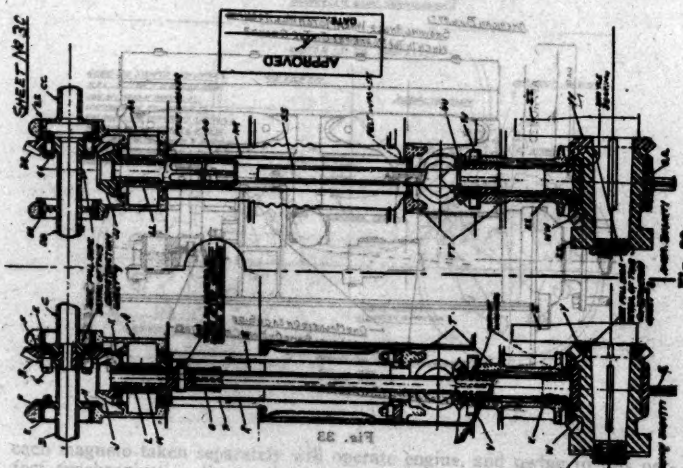



Fig. 30.

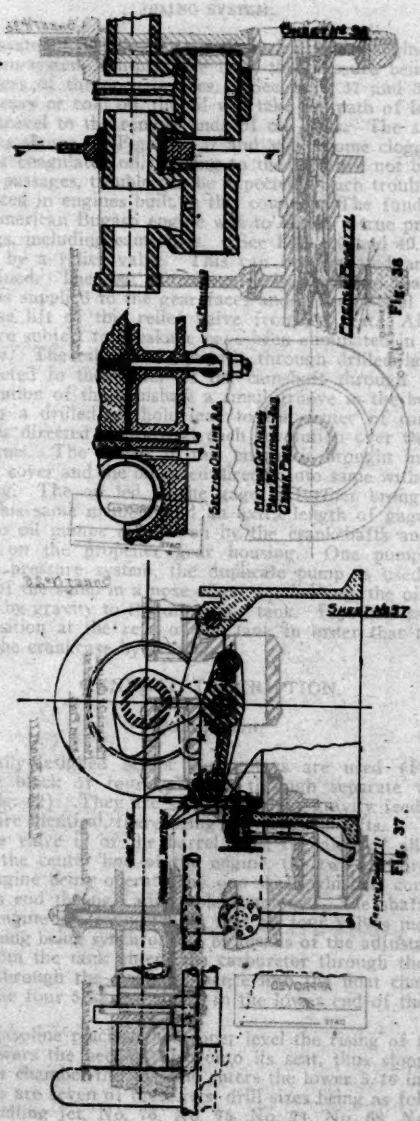


Fig. 36

Fig. 37

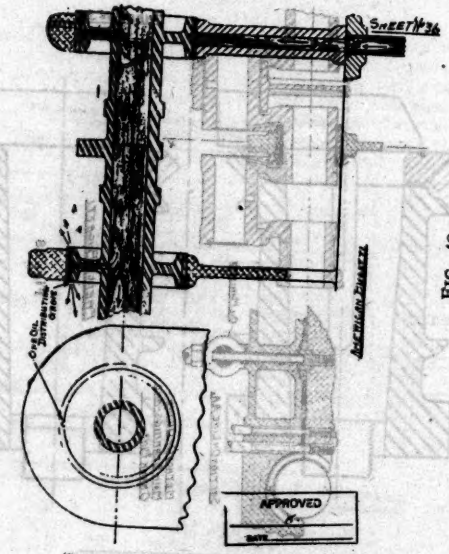



FIG. 40.

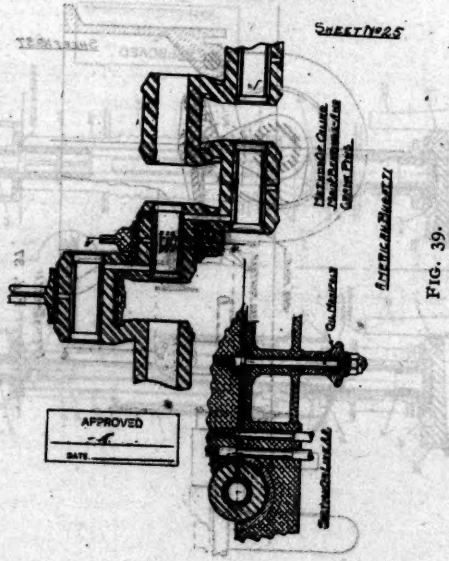



FIG. 39.

### OILING SYSTEM.

The oiling system as applied to the French Bugatti engine can be considered an open-system non-pressure type, the pressure being determined by the diameters of the open orifices. (See Figs. 37 and 38.) In other words, with heavy or cold oil, the oil will take the path of least resistance and will not travel to the remote ends of oil leads. The small openings in bearings (as shown on Figs. 36, 37 and 38) become clogged with sediment, waste or coagulated oil. Owing to the pressure not being sufficient to clear these passages, trouble can be expected. Such trouble has already been experienced in engines built in this country. The fundamental principle in the American Bugatti engine was to obtain a true pressure system on all bearings, including camshafts. (See Figs. 39 and 40.) The oil to be controlled by a relief valve. This can be regulated and the proper pressure obtained. The seat of this valve is scored so that at all times a flow of oil is supplied to the gear faces and an additional flow will also come from the lift of this relief valve from its seat. All exterior oil pipes which are subject to breakage have been eliminated in the American Bugatti engine. The oil is fed upwards through drilled holes in crankcase and directed to the center of the camshaft through two openings. At each revolution of the camshaft a small groove in the bushing of this bearing passes a drilled oil hole lead to the center of camshaft. This squirt of oil is directed outward at each revolution over the cams, gears and rocker arms. The tie-bar in this engine is brought in contact with the front gear cover and the oil is led directly into same without the use of exterior piping. The oil led to the gage is further brought out at the rear end of this same member and an extra length of gage pipe is thus avoided. Two oil pumps are driven by the crankshafts and said pumps are mounted on the propeller gear housing. One pump is arranged to supply the pressure system, the duplicate pump is used to clear the forward end of the sump in a nose dive. In climbing, the oil is led out of the crankcase by gravity to the oil supply tank. Pipe into said tank being led into a position at the rear of the tank in order that the oil cannot flow back in the crankcase by gravity.

## GENERAL DESCRIPTION.

### CARBURETION.

Four specially-designed Miller carburetors are used (Fig. 41), each supplying one block of four cylinders through separate water-jacketed manifolds (Fig. 42). They are set low so that gravity feed may be used and all four are identical, there being no rights or lefts.

The throttle valve is of the barrel type, the axis of all valves being parallel with the center line of the engine, the two carburetors on each side of the engine being operated by one shaft which is connected to the valves at each end through adjustable couplings. The shafts on the two sides of the engine are connected so that all four valves move in unison, the valve opening being synchronized by means of the adjustable couplings.

Gasoline from the tank enters the carburetor through the elbow (Fig. 41), passing through the strainer, thence into the float chamber, flowing out through the four 5/32 inch holes in the lower end of the needle-valve seat.

When the gasoline reaches the proper level the rising of the cylindrical brass float lowers the needle valve onto its seat, thus stopping the flow. From the float chamber the gasoline enters the lower 3/16 inch hole in jet holder. There are seven of these jets, drill sizes being as follows: No. 76, which is the idling jet, No. 76, No. 75, No. 71, No. 68, No. 57, No. 53. These jets progressively come into action as the throttle is opened.

Gasoline is drawn into the jet through the small hole in the bottom of the threaded end, mixing with a certain amount of air sucked in through the four holes drilled in the barrel of the jet just above the threaded portion. This air is taken from the outside through the upper 3/16 inch hole in jet holder and passes down around the outside of the jet to the four holes mentioned above. The major portion of the air enters the carburetor through the lower end of the venturi, which is 3 inches in diameter, passes up around the jet bar holder, combining above this with the rich mixture from the jets to form the proper mixture for combustion.

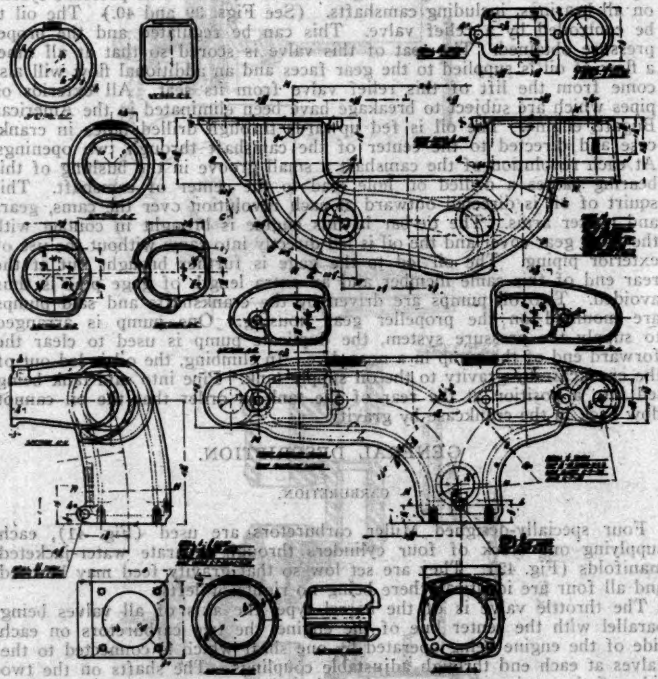


Fig. 42—Details of Gas Inlet Manifold—Front right and rear left.

Assembly of the altitude valve is shown on drawing (Fig. 43). This valve operates by turning the lever which is attached to the altitude control valve. This valve has two openings in its seat which when in the open position register with two similar openings in the stationary cover, thus making two free passages to the outer air, the size of this passage being governed by the position of the lever.

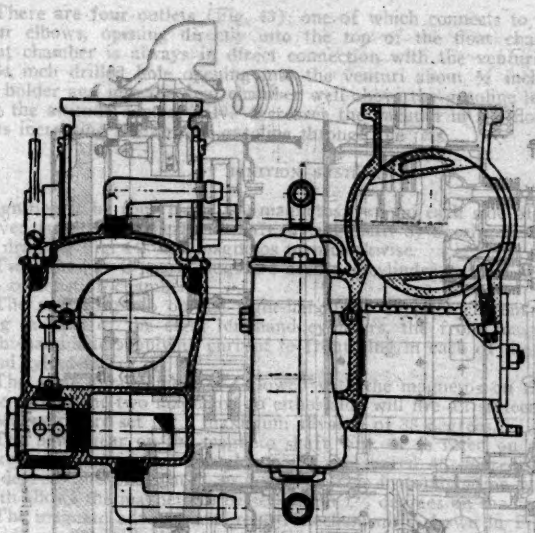


Figure 41—Carburetor Assembly

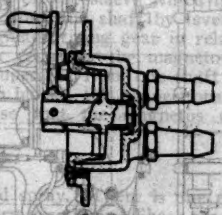


Figure 43—Altitude Control Valve Assembly

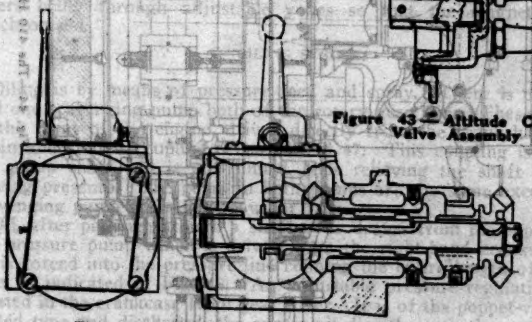


Figure 45—Magneto Gear Housing Assembly

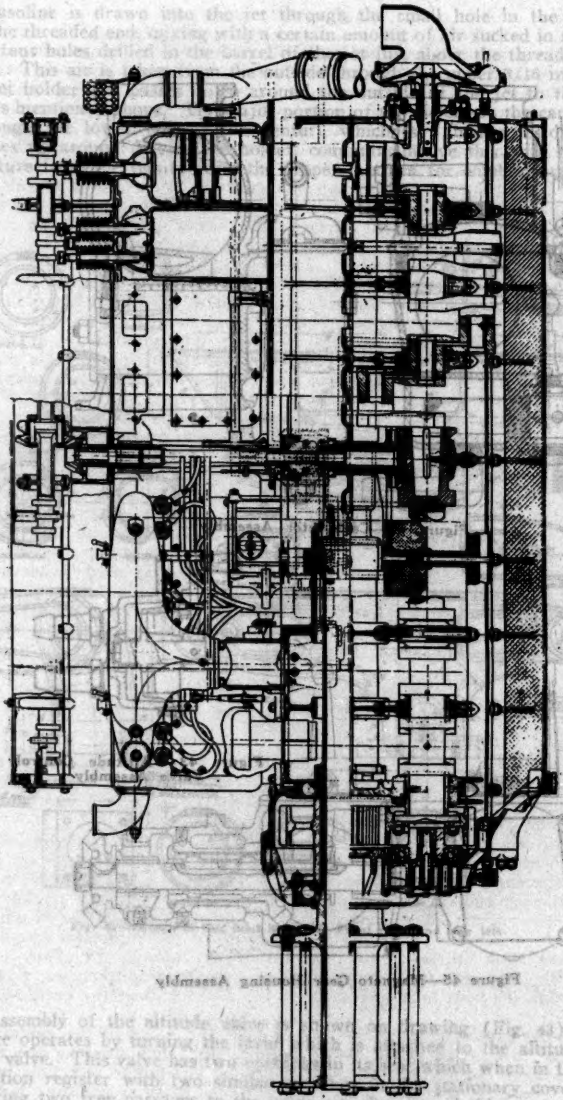


Fig. 44—The 410 H.P. King-Bugatti engine—longitudinal section.

There are four outlets (Fig. 43), one of which connects to each of the four elbows, opening directly into the top of the float chamber. The float chamber is always in direct connection with the venturi through a 5/64 inch drilled hole opening into the venturi about ¼ inch above the jet holder and into the float chamber well above the gasoline level. Opening the altitude control valve decreases the vacuum in the float chamber, thus increasing the flow of gasoline through the jets.

### IGNITION SYSTEM.

Ignition is by four Dixie 800 magnetos, two on each side of the engine, driven from the vertical camshaft driving shaft by bevel gearing as shown on drawing Fig. 44. All magnetos turn clockwise.

Two Titan A C spark plugs are used per cylinder located in the side of the combustion chamber.

The rear magneto on the right-hand side supplies current to the rear plug in each of the eight left-hand cylinders, the front magneto on the right-hand side supplying current to front plug in each of the eight-right-hand cylinders.

The same arrangement is followed with the magnetos on the left-hand side so that the two magnetos on either side will fire all sixteen cylinders.

Magnetos are set for a maximum advance of 38 degrees.

The bevel gear on the magneto shaft, Fig. 44, is fitted on a taper with a key. The gear has eight keyways, six spaced 48 degrees, one spaced 42 degrees and one spaced 30 degrees. This in combination with the gear teeth allows the magneto to be set within 1½ degrees on the crankshaft.

The magneto advance and retard mechanism is shown in Fig. 45. Gear inside meshes with the gears on the magneto shafts. This gear has four internal spiral grooves sliding over splines on the sleeve which is keyed to the driving shaft, but may be moved along the shaft by lever. The movement of this sleeve revolves the magneto-driving gear in relation to the shaft-driving gear, thus advancing or retarding the magnetos. The levers on the two sides of the engine are operated from one shaft located above the crankcase between the cylinder blocks, the connections to the levers being through adjustable yokes so that the magnetos may be synchronized.

### OILING SYSTEM.

Oiling is by means of pressure feed and spray. There is one pressure and one scavenging pump both of the rotary gear type. These are located at the front of the engine, driven directly from the crankshafts through a pin and slotted coupling, Figs. 46 and 47. This coupling is squared to the pump shaft, but is not pinned, thus relieving the shaft of any end driving pressure. The gears in both pumps are the same except that the scavenging pump gears have a wider face.

Oil, after passing through a strainer, is drawn from the supply tank by the pressure pump which is driven from the right-hand crankshaft. This oil is forced into the pressure line running the entire length of the crankcase as indicated in Fig. 48. An adjustable pressure-regulating valve is located in the crankcase front gear cover. It is of the poppet-valve spring-seated type and discharges the excess oil directly onto the propeller shaft driving gears. This valve is generally set so that the pressure gage which is connected to the rear end of the main oil line in the crankcase registers about 30 pounds. This valve has holes drilled through the head so that there is always a certain amount of oil discharged onto the gears.

From the pressure line the oil passes up around the studs which hold this line in position to an oil passage cut along the top surface of the crankshaft bearing cap, see Fig. 48. For the center crankshaft bearing

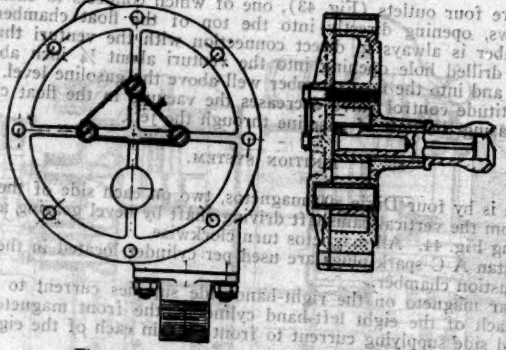


Figure 46—Oil Pressure Pump Assembly

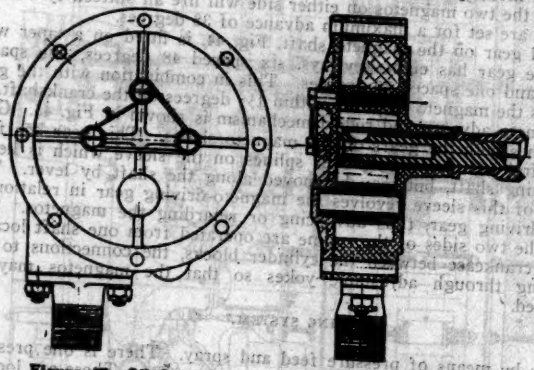


Figure 47—Oil Suction Pump Assembly

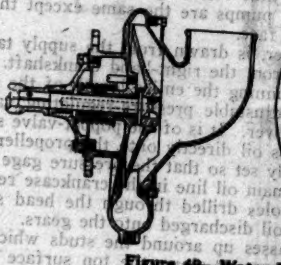


Figure 48—Water Pump Assembly

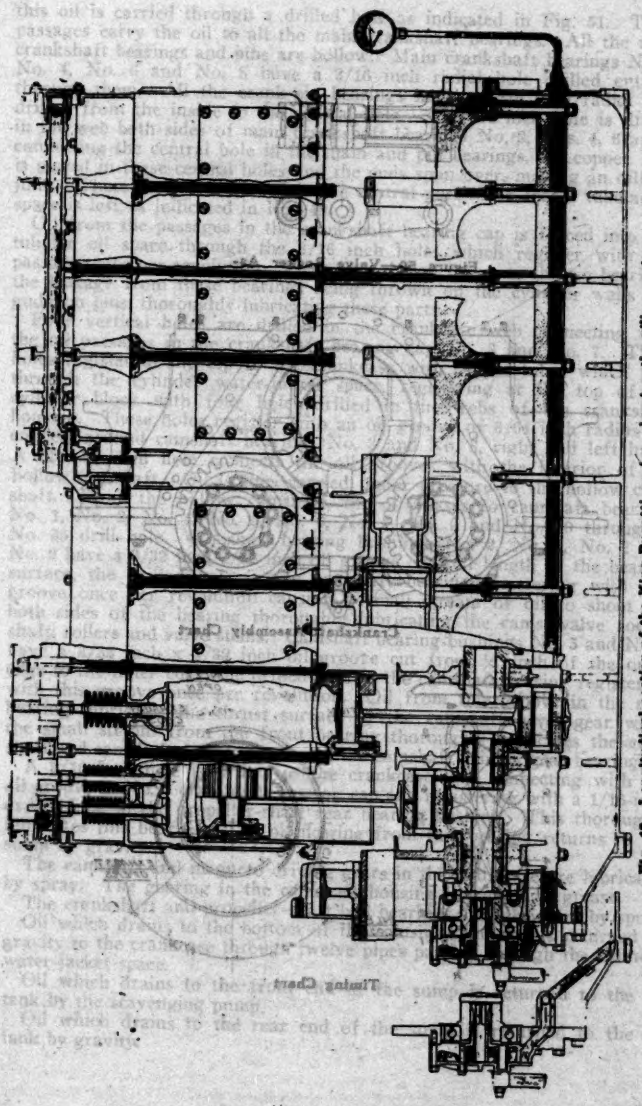


Fig. 48.—Longitudinal section of the King-Bugatti engine, showing the oiling system

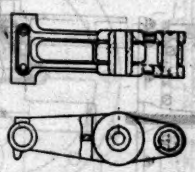
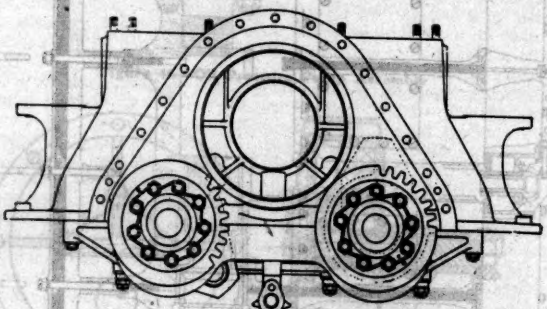
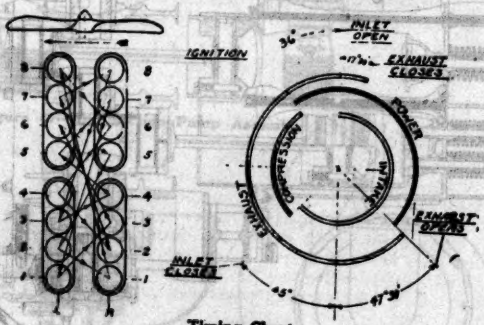


Figure 50—Valve Rocker Assembly



Crankshaft Assembly Chart




Timing Chart

this oil is carried through a drilled hole as indicated in Fig. 51. These passages carry the oil to all the main crankshaft bearings. All the main crankshaft bearings and pins are hollow. Main crankshaft bearings No. 2, No. 4, No. 6 and No. 8 have a 3/16 inch radial hole drilled entirely through them. All the crank-pin bearings have a 3/16 inch radial hole drilled from the inside to the central hole. A 13/64 inch hole is drilled in the web both sides of main crankshaft bearings No. 2, Nos. 4, 6 and 8 connecting the central hole in the main and pin bearings. A copper shell is placed in these central holes and the ends spun over, making an oiltight joint. These shells are necked in the central portion so that a tubular oil space is left as indicated in Fig. 48.

Oil from the passages in the crankshaft bearing cap is forced into this tubular oil space through the 3/16 inch holes which register with this passage twice per revolution. From here the oil passes to the pin bearings, the leakage from these bearings being thrown on the cylinder walls and gudgeon pins, thoroughly lubricating these parts.

Four vertical holes are drilled in the crankcase web connecting with the oil passages in the crankshaft bearing cap No. 3 and No. 7. These holes register at the top of the crankcase with copper tubes which pass through the cylinder water-jacket space, registering at the top of the cylinder block with four holes drilled in the webs of the crankshaft housing. These holes register with an oil groove of 3/64 inch radius cut entirely around camshaft bearings No. 3 and No. 8, right and left hand. A No. 35 drill hole connects the oil grooves with the interior of the hollow camshaft. Oil is thus carried under pressure to the hollow camshaft. From the hóllow camshaft, the oil passes to camshaft bearings No. 1, No. 2, No. 4, No. 5, No. 6, No. 7, No. 9 and No. 10 through a No. 35 drill hole. Camshaft bearing bushings No. 2, No. 4, No. 7 and No. 9 have a 3/32 inch 1/32 inch oil groove cut full length of the bearing surface, the drilled hole in the camshaft bearing registering with this groove once per revolution causing a small stream of oil to shoot out both sides of the bearing thoroughly lubricating the cams, valve rocher shaft, rollers and valve stems. Camshaft bearing bushings No. 5 and No. 6 have a 3/32 inch x 1/32 inch oil groove cut from ¼ inch of the outer edge to the inner edge, the drilled hole in the camshaft bearing registering with this groove once per revolution. Oil from this groove in the rear bearing lubricates the thrust surface of the camshaft bevel gear while the small stream from the front bearing thoroughly lubricates the camshaft and camshaft driving gears and the camshaft driving gear bearings.

A 3/16 inch hole is drilled in the crankcase web connecting with the oil groove in No. 6 crankshaft bearing cap and registering with a 1/16-inch drilled hole in the propeller-shaft rear bearing bushing. This thoroughly lubricates this bearing. The oil flowing from this bearing returns to the sump by gravity.

The camshaft and magneto driving gears in the crankcase are lubricated by spray. The gearing in the camshaft housing is packed in grease.

The crankshaft and propeller-shaft ball bearings are lubricated by spray.

Oil which drains to the bottom of the camshaft housing is returned by gravity to the crankcase through twelve pipes passing through the cylinder water-jacket space.

Oil which drains to the front end of the sump is returned to the oil tank by the scavenging pump.

Oil which drains to the rear end of the sump is returned to the oil tank by gravity.

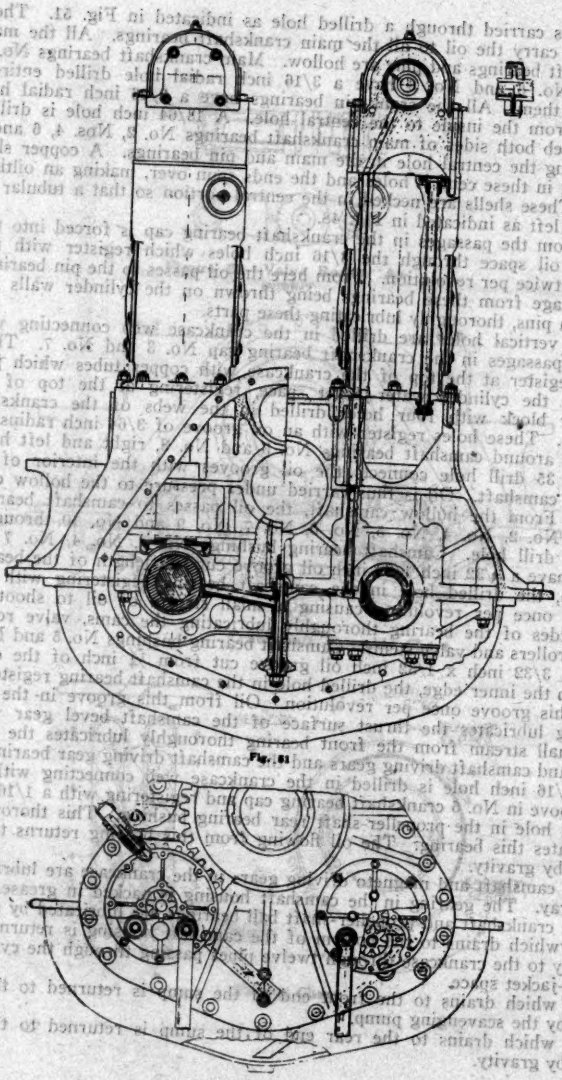


Fig. 81

Section showing oiling system—King-Bugatti Engine

## COOLING SYSTEM.

Cooling water is circulated through the engine by means of a centrifugal pump, see Fig. 49, driven from the rear end of the left-hand crankshaft by a pin and coupling the same as used on the oil pump.

The cooling system from the pump inlet to the outlet elbows on the front cylinders holds four and one-quarter gallons of water.

The pump impeller is 5⅞ inches diameter with eight vanes, the web being drilled with eight ⅜-inch holes on a circle of 2 inches diameter to equalize the water pressure.

The pump shaft is packed with a graphited asbestos rope packing, automatically held under compression by a coiled spring acting on the gland.

The pump shaft is hollow, the rear end being in direct communication with the water in the pump case. Water entering the shaft is forced out to the shaft rear bearing surface through a ⅛-inch hole. Any leakage of water past the asbestos packing is drained outside of the crankcase through a ⅜-inch cored hole in the water-pump body. The front bearing on the pump shaft is lubricated by spray from the crankcase which collects on the shaft bushing support and drains down into a ⅛-inch hole leading to the bearing. Any oil leakage from the front end of this bearing returns to the sump, any slight leakage from the rear end of the bearing is drained outside of the crankcase with the water leakage from the rear bearing.

There is one water inlet to the pump 2¼ inches inside diameter while the single outlet is 2 3/16 inches diameter. Water from the pump is forced up into an aluminum pipe with one branch leading to the rear end of each of the rear cylinder blocks, water entering the cylinders at the top of the water jacket on the exhaust side. A certain amount of the water circulates through the inlet manifold jacket, the remaining filling the cylinder water jacket space.

Cylinder blocks are cast with integral water jackets except the sides below the inlet and exhaust ports which are covered by an aluminum water jacket plate held in position by screws.

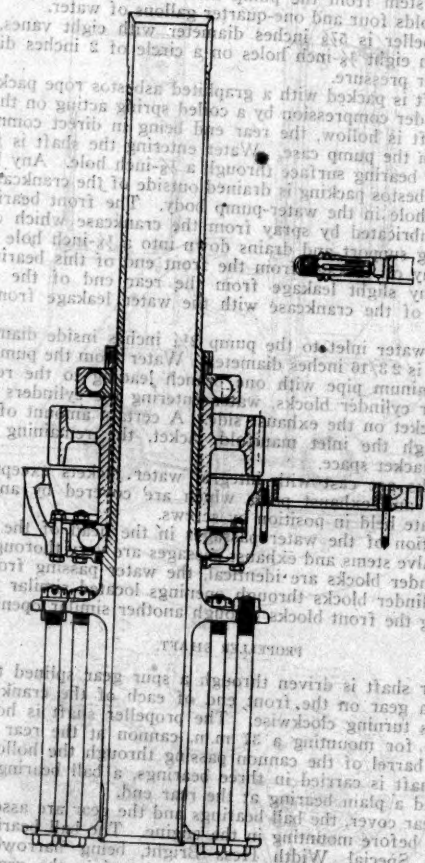
The construction of the water passages in the head of the cylinder is such that the valve stems and exhaust passages are very thoroughly cooled.

All four cylinder blocks are identical, the water passing from the rear to the front cylinder blocks through openings located similar to the inlet opening, leaving the front blocks through another similar opening.

## PROPELLER SHAFT.

The propeller shaft is driven through a spur gear splined to the shaft meshing with a gear on the front end of each of the crankshafts, both the crankshafts turning clockwise. The propeller shaft is hollow. Provision is made for mounting a 37 m.m. cannon at the rear end of the crankcase, the barrel of the cannon passing through the hollow propeller shaft. This shaft is carried in three bearings, a ball bearing either side of the gear and a plain bearing at the rear end.

The front gear cover, the ball bearings and the gear are assembled complete as a unit before mounting in the engine. The ball bearings are No. 6219 Monarch Special, Width Hess Bright, being narrower than the standard bearing. The front bearing is mounted in the gear cover and takes all the propeller thrust as well as a certain part of the radial load. The rear bearing slides into a retainer in the crankcase, being free to move endwise, carrying radial load only. The hub of the gear acts as a spacer for the ball bearings, they being held in position on the shaft by two nuts with a locking plate between. Mounting is such that the ball bearings and gear are easily and positively assembled there being no danger of injuring the ball bearings by screwing the retaining nuts too tight.


Propeller shaft and front cover assembly

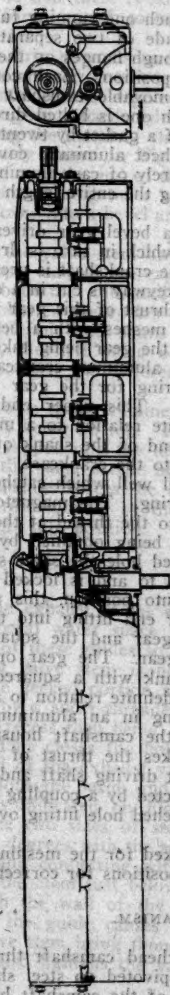
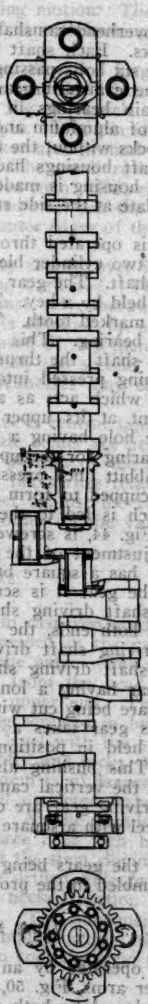


Fig. 83—Crankshaft Housing Assembly—Right



Crankshaft Assembly—Right hand

## CAMSHAFT DRIVE.

There are two overhead camshafts, each one running full length of the two cylinder blocks. Each shaft is made of two separate shafts joined near the center by six bolts passing through flanges on the shafts. These bolts also hold the camshaft gear in position. Each complete shaft is carried in ten plain bearings in a removable camshaft housing. The two housings are of aluminum and each one is bolted directly to the top of the cylinder blocks without the use of a gasket by twenty ¼-inch studs.

The first camshaft housings had a sheet aluminum cover as indicated on Fig. 51, a later housing is made entirely of cast aluminum with a cast aluminum cover plate at the side running the entire length of the housing. See Fig. 52.

Each camshaft is operated through a bevel gear driven by a vertcial shaft between the two cylinder blocks which in turn is driven by a bevel gear on the crankshaft. The gear on the crankshaft is pressed in position and in addition is held by a key. The keyway is cut in a definite position with relation to a marked tooth. The thrust of this gear is taken by the crankshaft center bearing. This gear meshes with a bevel gear which drives the vertical shaft; the thrust of the gear being taken on a bronze Babbitt lined bushing pressed into the aluminum crankcase. This gear has a long shank which acts as a bearing for the gear and has a fine external thread cut at its upper end. This upper end has a square broached hole, the hole having a definite relation to a marked tooth of the gear. The bearing for the upper end of the shank of this gear is a bronze bushing Babbitt lined pressed into the crankcase. The upper end of the bearing is cupped to form an oil well which catches the spray in the crankcase which is led to the bearing. The magneto driving shaft driving gear, see Fig. 44, is screwed into the thread at the upper end of the shank, the adjustment of the gear being obtained by means of the thread. This gear has a square broached hole the same size as the hole in the shank of the gear it is screwed to and is locked in position by dropping the camshaft driving shaft into position, this shaft having a squared section at both ends, the lower end fitting into the square hole in the magneto driving shaft driving gear and the square hole of the shank of the camshaft driving shaft gear. The gear on the camshaft meshes with a gear having a long shank with a squared section at its lower end, the square being cut with a definite relation to a marked tooth on the rear. This gear takes a bearing in an aluminum Babbitt-lined bushing which is held in position in the camshaft housing by a large flat-head screw. This bushing also takes the thrust of the bevel gear. The upper end of the vertical camshaft driving shaft and the lower end of the camshaft driving gear are connected by a coupling consisting of a square piece of steel with a square broached hole fitting over these square ends.

The teeth of all the gears being marked for the meshing position they may be easily assembled in the proper positions for correct timing.

## VALVE MECHANISM.

The valves are operated by an overhead camshaft through a rocker arm. These rocker arms, Fig. 50, are pivoted on steel shafts which are slid into drilled holes along both sides of the camshaft housing as indicated in Fig. 50. There are four of these rods to each housing, each one-half the length of the housing. The ends of the rods butt together at the center, the outer ends being flush with the ends of the camshaft housing and covered by the end bearing support for the camshaft when this is bolted in position. The rods are thus prevented from moving

lengthwise. They are of a tight enough fit in the drilled hole in the camshaft housing so that there is no turning motion. The outer end of each rod is tapped for a wrench to be used in withdrawing the rods from the housing.

Each rocker arm operates one valve, they are forgings and the pivot bearing of the arm on the rod in the camshaft housing is not bushed. The cam operates on the large roller, Fig. 50, which is of hardened steel taking a bearing on a hardened-steel pin. This pin is held in the rocker arm by spinning the metal of the rocker arm around the beveled end of the pin as clearly shown in Fig. 50. The small roller operates directly on a cap placed over the end of the valve stem. This roller is of hardened steel and takes a bearing on a hardened-steel pin the ends of which are soft and spun over into a bevel at the outer edges of the hole in the rocker arm.

The end of the valve stem has a cap slipped over it as is indicated in Fig. 44, the upper end of the cap being hardened. The proper clearance .015 inch for both the inlet and exhaust valve between the end of this cap and the roller in the rocker arm is obtained by placing shims in the cap. Three different steel shims are used of .003 inch, .005 inch and .010 inch thickness, the first being octagonal, the second hexagonal and the third round, so that the different thicknesses may easily be picked out by eye.

The upper valve-spring retainers have a central tapered hole, the large end of the hole being on top. The vale stems are necked and a tapered split collar is slipped into the necked portion of the stem, large end up. This taper is the same as the hole in the spring retainer. The pressure of the valve springs forces the retainer against the tapered collar which is prevented from moving by the shoulder on the valve stem thus locking the retainers into position.

### VALVES.

The inlet and exhaust valves work in cast-iron guides pressed into the cylinders. Liberal water space is provided in the cylinder head in the neighborhood of these guides so that the valve stems are well cooled.

The exhaust valve stem is hollow from the head to within a short distance of the necked portion at the upper end. The hole is closed at the head end by a short threaded plug screwed in below the surface of the valve, the recess then being filled level with the surface of the valve by welding. This closes the hole tightly and locks the plug in position. The lower end of the exhaust valve stem is of larger diameter than the upper end. Both the large and small diameters take a bearing in the valve guide as indicated on Fig. 44. At the shoulder formed by the junction of the two sizes of stem three 3/32 inch holes are drilled at an angle of 30 degrees with the axis of the stem sloping towards the head of the valve and connecting with the drilled hole in the stem. At the upper end of the stem just below the necked portion a 5/32 inch hole is drilled through the wall of the stem. The movement of the valve up and down in the guide causes a pumping action, the transfer of air within the valve stem being thought to cool the stem to a certain extent. This drilling also lightens the valve.

### PISTON.

The piston is of aluminum and has two ring grooves above the gudgeon pin. Two ⅛-inch wide rings are placed in both of these grooves. The rings require a pressure of ten to twelve pounds applied on a diameter

at right angles to the slot to bring the ends of the 30-degree slot to within .010 inch of the closed position. The lower ring is beveled, being placed in the groove with the sharp edge down so that it acts as a wipe ring forcing the oil on the down stroke into an oil groove cut just below the ring groove. The land between the bottom of the ring groove and the oil groove is 1/64 inch wide and it is 1/32 inch smaller diameter than the land above this ring groove. This forms a free passage for the oil to the oil collecting groove. Eight 3/32-inch holes are drilled around the piston connecting the oil groove with the interior of the piston. These holes slope down at an angle of 60 degrees with the axis of the piston carrying the oil wiped from the cylinder wall to the interior of the piston. A slot ⅛ inch wide x 1/16 inch deep is cut from the oil groove to the gudgeon pin hole at both ends thus lubricating these bearings. Two ½-inch wide x 1/32-inch deep oil distributing grooves are cut around the piston, one about 3/16 inch above and one about 5/16 inch below the center line of the gudgeon pin.

GUDGEON PIN.

The gudgeon pin floats in both the connecting-rod bushing and the piston. Bearing in the piston is directly on the aluminum. It is held from endwise motion by an aluminum plug pressed into each end. See Fig. 44. These plugs are drilled axially with a 3/16-inch hole, which allows a certain amount of oil from the cylinder wall to enter the hollow gudgeon pin. The ends of these plugs where they bear against the cylinder wall are turned to a spherical seat of a radius equal to the radium of the cylinder bore. In operation the gudeon pins turn more or less. Two ⅛-inch holes, in line, one ¾ inch each side of the center of the pin, are drilled through the wall of the pin. These holes when in a certain position register with an oil groove in the connecting-rod bushing allowing a certain amount of oil to enter the hollowed gudgeon pin and when in the lower position allow the oil which is pocketed in the pin to run out onto the connecting-rod bearing.

CONNECTING ROD.

The connecting rod is a steel forging machined all over. It is of eye cross-section.

The small end bearing is a bronze bushing pressed in position, having one straight oil groove, 3/32 inch x 1/32 inch deep running to with ¼ inch of the ends of the bushing. This groove is in communication with a 13/32-inch drilled hole in the connecting rod through a ⅛-inch hole. Oil collects in the pocket formed by the 13/32-inch hole, and is led to the bearing through the ⅛-inch hole.

The big end of the connecting rod is fitted with a bronze bushing which is babbitt line. This end is split at right angles to the rod on the center line of the bearing, the cap being held in position by two ⅜-inch chrome nickel-steel bolts. The bushing is relieved at the line by 5 grooves, 3/16 inch wide, about ¾ inch long, 3/64 inch deep, at the parting line cut on a 7/16-inch radius with a center on the parting line. The rod half of the bushing has no oil groove, the cap half has a circular oil groove entirely around it cut on the center line of the bushing and registering with the oil-feed hole in the crank pin. This groove is ⅛ inch wide x 1/32 inch deep. The bushing in the cap is kept from turning by a dowel pin with an enlarged head. This head enters a countersunk hole in the cap, the small end entering the hole in the bushing when it is placed into position. This end is of such a length that it does not project through the bushing. The pin is thus locked in position and cannot drop out or rub on the shaft.

## CRANKSHAFT.

The crankshaft is made in two pieces connected at the center by a taper and key drawn up with a nut. Each section of the shaft forms a four-cylinder shaft with the throws all in one plane, the throws of the two sections being assembled at right angles. In assembling, the rear end of the front half is immersed in boiling water, the tapering end of the rear section which is cold is then slipped into position and the parts drawn together by the nut using a long handled wrench.

The rear end of each complete shaft has clutch teeth cut on it for attaching a starter.

All end thrust coming on the shaft is taken by the center or No. 5 bearing all the other bearings having about 1/16-inch clearance at both ends.

All bearings, including the connecting rod bearings, with the exception of the center main bearing are undercut. This results in a total shortening of the shaft of approximately 4 31/32 inches. This of course results in a considerable saving in the weight of various parts while still allowing ample bearing surface.

The crankshaft main bearings are bronze bushing, babbitt lined. These bearings are not relieved at the parting line and there are no oil grooves excepting in the lower half of bearings No. 1, No. 2, No. 3, No. 4, No. 6, No. 7, No. 8. These have a 3/32-inch wide x 1/32-inch deep circular oil groove entirely around them. This groove registers with the oil into the hollow crankshaft bearing. As this oil hole is drilled through both walls of the bearing on a diameter and as the oil groove in the bushing extends through 180 degrees there is always a free passage for the oil from the bearing into the hollow crankshaft.

In assembling the completed crankshafts in the crankcase, they are placed in such a relation to each other that if No. 8 throw left is on top dead center No. 8 throw right will be 45 degrees past bottom dead center. Both cranks turn clockwise viewed from the rear of the engine.

The propeller driving gears are bolted to the crankshaft with nine bolts equally spaced, the bolt holes being drilled in a certain relation to the gear teeth. This makes it possible to use the same gear on either shaft with a maximum error in the setting of the shafts of 20 minutes. The flange on the crankshaft and the gears are marked as indicated in Fig. 51 for the proper position of assembly of the gears on the shafts.

## CYLINDERS.

The cylinders are of iron cast in blocks of four. They are bolted directly to the top of the crankcase without the use of a gasket. The water jacket is cast integral with the exception of the sides of the cylinder block below the inlet and exhaust ports, which are covered with a cast aluminum plate attached with screws. A gasket is used between these plates and the cylinder.

Cylinders are cast with separate exhaust ports. One inlet port supplies two cylinders. Two inlet and one exhaust valves are used.

The entire combustion space is machined with the exception of a very small recess near the inlet valve seat.

The tops of the cylinder blocks are machined, making an oiltight joint with the camshaft housing without the use of a gasket.

Provision is made for a liberal circulation of water in the neighborhood of the valve ports, seats and guides overcoming valve trouble.

Spark plugs, of which there are two per cylinder, are located at the side of the combustion chamber in close proximity to the inlet valves and are well cooled by the circulating water.

### CRANKCASE.

The crankcase and oil pan are of cast aluminum. The case is well ribbed. All bearings are supported from the upper part of the case, the bearing bushings being held in place by caps which extend almost the full width of the case. Each cap supports two bearings. They are all of cast aluminum with the exception of the center bearing cap, which takes all the thrust of the crankshaft. This is a steel forging.

The oil pan has a web running its entire length along the center line of the engine. This greatly assists in preventing the current of air caused by the revolving crankshafts from drawing up the oil which is constantly draining into the pan. There is also a cross web between each cylinder block which runs well up the sides of the oil pan. These cross webs have an opening at the bottom at their center, allowing the oil to drain to the ends of the pan on either side of the web running lengthwise.

A breather is attached to the top of the crankcase near the front end between the cylinder blocks.

Provision is made at the rear end of the case for attaching a 37 m.m. cannon, the barrel of the cannon projecting through the hollow propeller shaft.

### GUN CONTROL.

Provision is made for attaching two gun-control mechanisms, one at the rear end of each of the camshaft housings driven directly from the camshafts through a slotted coupling. When this control mechanism is used it is inserted in place of the camshaft housing end bearing cover.

### TACHOMETER DRIVE.

The tachometer drive may be taken from either end of the two camshafts. It operates at camshaft speed through a slotted coupling as indicated in the illustration.

When the gun control mechanism is used the tachometer drive is taken from the rear end through the same type of coupling as when driven directly from the end of the camshaft.

### GENERAL DATA

| | |
|---|---|
| Number and arrangement of cylinders, | 16 vertical, 2 rows of 8 in blocks of 4 |
| Material | Cast iron. |
| Bore | 4.33" 110 m.m. |
| Stroke | 6.3" 160 m.m. |
| Stroke-bore ratio | 1.455 : 1 |
| Area of one piston | 14.725 sq. in. |
| Total piston area | 235.6 cu. in. |
| Swept volume of one cylinder | 92.768 cu. in. |
| Displacement of motor | 1484.288 cu. in. |
| Compression ratio | 5 : 1 |
| Normal brake H.P. | 410 at 2,000 r.p.m. |
| Type of valve gear | Overhead camshaft and valve rockers. |
| Number of carburetors | Four Miller special. |
| Ratio propeller to crankshaft speed | .666 : 1 |

### VALVES.

| | |
|---|---|
| Number per cylinder | Two inlet and one exhaust. |
| Outside diameter, inlet | 1.535 |

| | |
|---|---|
| Outside diameter, exhaust | 2.263 |
| Port diameter, inlet | 1 27/64" |
| Port diameter, exhaust | 2 3/64" |
| Width of seat, inlet | .050" |
| Width of seat, exhaust | .108" |
| Angle of seat | 10° |
| Valve lift, inlet | .653" |
| Valve lift, exhaust | .700" |
| Diameter of stem, inlet | .352" |
| Diameter of stem, exhaust (large) | .592" |
| Diameter of stem, exhaust (small) | .4355" |
| Length of valve, inlet | 5 17/64" |
| Length of valve, exhaust | 5 13/32" |
| Number of springs per valve, inlet and exhaust | 2 concentric. |
| Length of spring in position, inlet (small) | 2 7/64" |
| Length of spring in position, inlet (large) | 2 15/64" |
| Length of spring in position, exhaust (small) | 2 27/64" |
| Length of spring in position, exhaust (large) | 2 27/64" |
| Mean diameter of coils, inlet spring (small) | 57/64" |
| Mean diameter of coils, inlet spring (large) | 1 9/32" |
| Mean diameter of coils, exhaust spring (small) | 1 9/32" |
| Mean diameter of coils, exhaust spring (large) | 1 23/32" |
| Clearance inlet valve stem | .015" |
| Clearance exhaust valve stem | .015" |
| Valves parallel to center line of cylinder bore. | |

CYLINDERS.

| | |
|---|---|
| Overall height of cylinders | 10 27/64" |
| Length of projection in crankcase | 3 1/16" |
| Width of cylinder casting at head over water jacket space | 5 7/16" |
| Width of cylinder casting at barrel over water jacket space | 3 1/16" |
| Length of cylinder casting over water jacket | 19 5/32" |
| Thickness of flange (base) | 7/16" |
| Number of studs per block of four cylinders | 20 |
| Diameter of stud | 5/16" |
| Thickness of water jacket wall side and head | 5/32" |
| Thickness of combustion chamber wall | 13/64" |
| Thickness of cylinder barrel, above flange of water jacket | 3/16" |
| Thickness of cylinder barrel, above flange below water jacket | 1/32" |
| Thickness of cylinder barrel below flange | 9/64" |
| Thickness of valve ports | 5/32" |
| Diameter of port at valve, inlet | 1 27/64" |
| Diameter of port at valve, exhaust | 2 3/64" |
| Inlet port at flange (for two cylinders) | 2 9/32" x 1¾" |
| Exhaust port at flange (for one cylinder) | 2 3/64" x 1 31/64" |
| Number of spark plugs per cylinder | 2 |

PISTON.

| | |
|---|---|
| Type of piston | Crowned. |
| Material | Aluminum. |
| Length of piston | 4 1/16" |
| Length to diameter ratio | .938 : 1 |
| Number of rings per piston; placed two in a groove lower ring beveled acting as wipe ring | 4 |
| Position of rings | Above gudgeon pin. |
| Width of rings | 1/8" |

| | |
|---|---|
| Width of land | ¼" |
| Distance from bottom to center of gudgeon pin | 2 1/16" |
| Thickness of head at center | ¼" |
| Thickness of head at edge | ⅜" |
| Thickness of wall at bottom | ⅛" |

GUDGEON PIN.

| | |
|---|---|
| Diameter of gudgeon pin | 1 5/64" |
| Thickness of wall | .164 |

CONNECTING ROD.

| | |
|---|---|
| Type | Plain. |
| Length between centers | 10 7/16" |
| Ratio length to crank throw | 3.313 : 1 |
| Small end bearing | Bronze Bushing. |
| Outside diameter of bushing | 1 3/16" |
| Length of bushing | 2 3/32" |
| Length of small end of rod | 2 3/32" |
| Outside diameter small end rod at end | 1 11/32" |
| Outside diameter small end rod at center | 1½" |
| Type of section | Eye. |
| Depth (small end) | 1⅛" |
| Depth (large end) | 1 7/32" |
| Width | ⅝" |
| Thickness of web | 5/32" |
| Thickness of flange (small end) | 5/32" |
| Thickness of flange (large end) | 13/64" |
| Large end bearing | Bronze babbit lined. |
| Inside diameter bushing | 2 3/16" |
| Outside diameter bushing | 2⅜" |
| Length | 2 9/64" |
| Thickness of babbit | 3/64" |

CRANKSHAFT.

| | |
|---|---|
| Number of crankshafts | 2 |
| Number of bearings (Plain) per crankshaft | 9 |
| Number of bearings (Ball) per crankshaft | 1 |
| Cylinder centers (in block) | 4 17/32" |
| Cylinder centers (between blocks) | 7 15/32" |
| Crank pins, outside diameter | 2 3/16" |
| Inside diameter | 1⅛" |
| Length | 2 11/64" |
| Main Bearings: | |
| Outside diameter Nos. 1, 2, 3, 4, 6, 7, 8, 9 | 2 3/16" |
| Outside diameter No. 5 | 2⅜" |
| Inside diameter, Nos. 1, 2, 3, 4, 6, 7, 8, 9 | 1⅛" |
| Length, Nos. 1, 2, 3, 4, 6, 7, 8 Bearing Bushings | 1 9/16" |
| Length No. 9 Bearing Bushing | 2 19/32" |
| Length No. 5 Bearing Bushing | 1 31/32" |
| Ball Bearing | Hess-Bright Monarch No. 6211 |
| Crank Webs: | |
| Width | 3 17/32" |
| Thickness | 43/64" |
| Radius of fillets | 3/32" |
| Weight of one complete shaft, for 8 cylinders, with propeller drive gear bolts and nuts, bevel gear and oil passage shells | 96½ lbs. |

### CAMSHAFT.

| | |
|---|---|
| Diameter of shaft | 1" |
| Inside diameter | 11/16" |
| Number of bearings | 10 |
| Diameter of Bearings Nos. 1 and 10 | 1" |
| Diameter of Bearings, Nos. 2, 3, 4, 5, 6, 7, 8, 9 | 2¼" |
| Length of Bearings Nos. 1 and 10 | 1 5/16" |
| Length of Bearings Nos. 2, 4, 7, 9 | ⅝" |
| Length of Bearings Nos. 3 and 8 | ¾" |
| Length of Bearings Nos. 5 and 6 | 27/32" |
| Width of cam face | 5/16" |
| Number of cams per cylinder | 3 |

### CAMSHAFT BEVEL GEAR.

| | |
|---|---|
| Pitch diameter | 3¾" |
| Number of teeth | 30 |
| Pitch | 8 |
| Width of face | ¾" |
| Diameter of bolt circle | 1 23/32" |
| Number of bolts | 6 |
| Diameter of bolts | 5/16" |

### CAMSHAFT HOUSING.

| | |
|---|---|
| Material | Aluminum. |

### CAMSHAFT DRIVING SHAFT.

| | |
|---|---|
| Diameter | ¾" |
| Wall thickness | 1/16" |

### CRANKCASE UPPER HALF.

| | |
|---|---|
| Material | Aluminum. |
| Thickness of wall | 3/16" to 5/16" |
| Thickness of supporting flange | 3/8" to 7/16" |
| Center distance of motor support bolts | 24¼" |
| Number of motor support bolts | 12 |
| Diameter of motor support bolts | ⅜" |
| Center to center of crank shafts | 10¼" |
| Height of case | 9.055" |

### CRANKCASE LOWER HALF.

| | |
|---|---|
| Material | Aluminum. |
| Thickness of wall | 3/16" |

### LUBRICATION.

| | |
|---|---|
| Type | Forced feed and spray. |
| Type of pump | Rotary Gear. |
| Number of pumps | One pressure, one scavenging. |
| Oil pressure | 30 lb. |

### PRESSURE PUMP.

| | |
|---|---|
| Number of teeth | 7 |
| Pitch | 6 |
| Outside diameter | 1½" |
| Width of face | 13/16" |
| Ratio of pump speed to crank shaft speed | 1 : 1 |

### SCAVENGING PUMP.

| | |
|---|---|
| Number of teeth | 7 |
| Pitch | 6 |
| Outside diameter | 1½" |
| Width of face | 1¼" |
| Ratio of pump speed to crank shaft speed | 1 : 1 |

### IGNITION.

| | |
|---|---|
| Type | Magneto. |
| Number | 1 |
| Make | "Dixie 800" |
| Firing order 1L, 7R, 3L, 4R, 3L, 8R, 7L, 2R, 4L, 6R, 8L, 1R, 2L, 5R, 6L, 3R | |
| Number of plugs per cylinder | 2 |
| Type of plug | Titan A. C. |
| Advance | 36° |
| Magneto Rotation | Clockwise. |

### COOLING SYSTEMS.

| | |
|---|---|
| Type | Water Cooled. |
| Pump | 1 Centrifugal. |
| Inside diameter of inlet and outlet elbow to cylinders | 1½" |
| Number of inlets | 2 |
| Number of outlets | 2 |
| Water temperature inlet | 150° F. |
| Water temperature outlet | 160° F. |

### WATER PUMP.

| | |
|---|---|
| Material | Aluminum. |
| Inside diameter of inlet | 2¼" |
| Inside diameter of outlet | 2 3/16" |
| Diameter of impeller | 5⅞" |
| Number of blades | 8 |
| Ratio pump speed to crankshaft speed | 1 : 1 |

### REDUCTION GEARS.

| | |
|---|---|
| Crank shaft propeller drive gear: | |
| Pitch diameter | 5.6" |
| Pitch | 5 |
| Number of teeth | 28 |
| Width face | 2⅛" |
| Diameter bolt circle | 3 17/32" |
| Number of bolts | 6 |
| Diameter of bolts | 7/16" |
| Propeller shaft gear | |
| Pitch diameter | 5.4" |

| | |
|---|---|
| Pitch | .5 |
| Number of teeth | 42 |
| Width face | 2¼" |
| Number of splines | 8 |
| Width of splines | .405 |
| Height of splines | .094 |

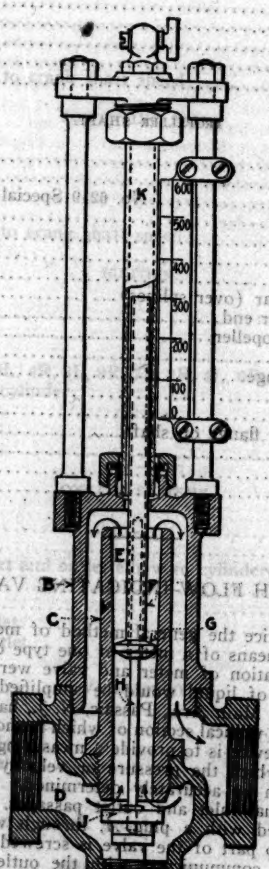
PROPELLER SHAFT.

| | |
|---|---|
| Number of bearings | 3 |
| Type of bearings | Two ball, one plain. |
| Ball Bearings | No. 6219 Special Width, Hess-Bright. |
| Plain bearing | Bronze babbitt lined. |
| Diameter of bearing | 3 7/16" |
| Length of bearing | 1½" |
| Thickness of babbitt | 1/32" |
| Outside diameter at gear (over splines) | 3¼" |
| Outside diameter at rear end | 3 5/16" |
| Outside diameter at propeller | 3 31/64" |
| Inside diameter | 2 61/64" |
| Diameter propeller flange | 9¾" |
| Thickness flange | 7/16" |
| Thickness loose flange | 9/32" |
| length of bearing loose flange on shaft | ¾" |
| Diameter of bolt circle | 7⅛" |
| Diameter of bolts | ⅝" |
| Number of bolts | 8 |

## EDLICH FLOW-INDICATING VALVE.

In power-plant practice the general method of measuring fluid passing through a pipe is by means of a meter of one type or another. It is evident that if a combination of meter and valve were made, the pressure or velocity indication of liquid would be simplified. With that idea in mind, Paul Edlich, 134 Main Ave., Passaic, N. J., has patented the Edlich flow-indicating valve, a vertical section of which is shown in the illustration.

The object of this device is to provide a measuring and indicating apparatus for liquids in which the pressure or velocity of the fluid passing through the valve can be accurately determined. The valve casting is provided with the usual inlet and outlet passages. The bottom of the valve body is provided with a plug *A*, through which the valve stem operates. Into the top part of the valve is screwed a hollow cylinder *B*, the interior of which communicates with the outlet of the valve body. Located within the hollow cylinder is an inverted conical-shaped body *C*, the lower end of which opens to the inlet passage *D* in the valve. The lower end of this conical body is so formed as to provide a valve seat. The body *C* is made with an inverted conical-shaped passage *E*, in which is mounted a sliding tubular indicating element *F*, the lower end of which is provided with a piston head *G*, which has a central opening that communicates with the interior of the indicating element, and extending through the opening in the piston head *G* and into the tubular indicating element *F* is the upper end of a guide rod *H*, the lower end of which is inserted into a valve disk *J* and which is operated by means of the hand-wheel below the plug *A*. A gage glass *K* is placed above the upper end of the hollow body *B*, with an adjoining graduated gage plate, as shown.




SECTION THROUGH IN-
DICATING VALVE

When the valve disc *J* is closed, the indicating element *F* will be at its lowest position, or zero. When the valve disc is moved to the open position, a pressure of the fluid against the lower face of the piston head *G* will force the piston and the indicating element *F* upward in the gage glass, and as the pressure of the liquid increases, the piston head and the indicating element will move upward in proportion to the amount of pressure, and by virtue of the graduations on the plate the pressure of the liquid against the piston head can be determined.

The liquid, in passing through the conical-shaped passage *E*, is discharged through the upper end of the passage into the space between the body *C* and the inner surface of the cylinder body *B*, as indicated by the arrows, and is then discharged through the discharge passage in the valve body.

## THE DIFFICULTIES OF WELDING STEEL BY THE OXY-ACETYLENE PROCESS.*

By B. K. Smith.

The question of steel welding is one of the most important of all the problems which confront the oxy-acetylene welder. A large part of this problem has to do with his preliminary knowledge of steels and his primary habits of handling the blowpipe in steel welding. The steel weld in itself is apparently one of the most easy to obtain, so that some welders are considering steel welding as a subsidiary problem of their trade because their welds look so well; yet steel welding, notably in boiler work, requires the most thought and care, and should be studied not only from the practical point of view, but also theoretically. And what are the difficulties of steel welding? They are: (1) The incorporation of iron oxide; (2) the thermal disturbance in the vicinity of the weld; and (3) the expansion and contraction problems. Therefore one needs to study: (1) The properties of various steels; (2) the phenomena produced during the melting of the metal under the oxy-acetylene torch; and (3) boiler construction.

If one has the above knowledge in a sufficient degree, the welding in itself will be then comparatively easy and successful; but if he has not, and failures occur to him, he will insist that certain operations are unpractical or impossible to perform with the oxy-acetylene blowpipe. This is one of the reasons why certain companies will not trust the apparently good-looking welds, and reserve to themselves the right to sanction or forbid this kind of welding for boiler or pressure work.

In studying the incorporation of iron oxide in the weld, one must know what iron oxide is and what it looks like. Iron oxide is a burnt element, created on the surface of molten steel under the action of the blowpipe and the atmospheric oxygen. It appears as white veins and foaming spots. Iron oxide will dissolve into the molten steel at the rate of a little higher than 1 per cent, and in this reaction it may destroy the carbon elements, with the result of a decrease in the strength and elasticity of the weld.

The best remedy to reduce oxidation is to reduce the oxygen pressure as low as possible—just enough to produce a free and soft flame that will produce a continuous melting without running over the edge. This operation depends largely on one's blow-pipe and the flame maintained throughout the weld. With a proper flame, one will notice the flow of the metal clean and regular, but with a too rigid flame which requires more

*Abstract of a paper read before the North-Western Welding Association, Minneapolis, Minn.

oxygen one will notice white veins and foamy spots flowing in the molten metal. These are streaks of oxide of iron, and they will dissolve and create new ones with the progress of the weld. The molten metal is swept rather than laid in clean layers, and in some cases the metal is adhered instead of welded. Oxide of iron cannot be altogether eliminated from any kind of weld so long as we cannot protect them from the effects of the atmosphere, but it can be so decreased from the interior of the weld that its contents would do no harm to the strength of the weld.

Some good and experienced welders who have been welding for five or six years, but who have met with some failures in steel welding, will not admit that the main cause of failure is the improper regulation of the flame, because to do so would be the same as for a college student to be compelled to return to the grades to learn his A B C's. Most of our welders have previously mastered some other craft and have begun welding on important work without any preliminary instruction. In other words, they began at the college end without having acquired the necessary underlying primary and high-school education.

What would a boilermaker think if a novice tried to put in a set of flues? Or a machinist if an apprentice attempted to make a set of dies? Yet the same boilermaker or machinist looks upon the welding blowpipe as merely a tool of his trade, and thinks that all he needs to do is to go ahead and weld with it.

In studying the question of the thermal disturbance produced by the gas flame in the metal when it is raised to a fusing temperature we must not forget that it is heat which has led to the creation of many new steels, and while heat can be employed to increase the life of metals it can also reduce and destroy the life and strength of metals. Therefore, it remains for us to study carefully how we shall best make use of it.

It is a well-known fact that the structure of any gas weld at the welding line, on a boiler plate, for instance, is cast metal. This metal is much lower in elongation than the boiler plate. Yet the structure of the boiler plate was originally at the mill a cast metal or something similar to the metal in the weld. But it has been refined by the operations of heat and mechanical treatment to a strong metal that we call boiler plate. The question is this: Can we employ the heating agent with the aid of the gas blowpipe to refine the grain at the weld and its vicinity? My answer is, "Yes," provided the operator knows the physical and the chemical properties of boiler steel.

There is no such thing as not being able to take care of expansion and contraction in steel welding. But in boiler welding one will find this a problem which is difficult to solve unless one has a full knowledge of the construction of boilers.

The effects of expansion and contraction are little feared by some boiler welders, because the metal to be welded possesses the property of elongation. My advice is that no welder should depend upon the metal giving, thinking that a little strain will not hurt the weld, for two reasons. One reason is that in almost all cases an intelligent welder can find a way to take care of the expansion and contraction. The second reason is that no human being can ever measure or know how much of a strain he has left in the metal. It may be very little or it may be up to the breaking point. Such a weld may crack during its own progress or a few days after, or even six months after completion; at any rate, the welder is not excusable, and the failure will not only condemn the operator, but the process and the whole oxy-acetylene welding industry.

There is one point that an oxy-acetylene operator must bear in mind, and this is to make a strong weld. Economy should be considered as a secondary principle. While there must not be any useless waste in time or material, yet we must not forget that a defective weld is a waste of time and material.—"Mechanical World."

## ELECTRICALLY WELDED STEAM LINES.*

By H. R. Woodrow, Assistant Chief Electrical Engineer, New York Edison Co.

In the construction of the United States Government Explosives Plant "C" at Nitro, W. Va., something over ten miles of steam lines carrying 190 pounds pressure were required. Of this about five miles were 12 inches diameter or over. At the time the construction of this plant was undertaken, in the spring of 1918, the available supply of large extra-heavy flanges was very small and it was doubtful if a sufficient quantity could be obtained within the time available for completing the plant. Therefore in order to finish the work in time the Thomas E. Murray Engineering Corp., which was handling the power-section engineering work, recommended, and the War Department adopted, the use of arc welding for the connection of all the high-pressure steam and feed-water lines 6 inches in diameter and over in the plant. At the time the armistice was signed over one thousand of these joints had been completed and were in service.

All the outside mains are suspended from timber bents, at a height of approximately ten feet about the ground, as shown in Fig. 3. The welding operation was performed directly under the bents and the section of welded-joint pipe, after being tested, was raised in place by means of chain blocks.

FIGS. 1 TO 3.—METHOD OF WELDING JOINTS AND INSTALLING STEAM LINE.

The length of welded-joint pipe so handled averaged approximately 200 feet and contained ten sections of mill-length pipe and nine welded joints, the two end sections having a standard Van Stone flange for connecting with fittings or valves in the completed line.

The type of welded joint used is a combination butt and sleeve joint, as indicated in Figs. 1 and 2. The butt ends of adjoining sections were scarfed off at an angle of about 45 degrees with the axis of the pipe and butt-welded. The weld was then chipped flush with the pipe surface, and a hydrostatic test of from 400 to 600 pounds was applied to the completed length. After inspection, the reinforcing sleeve was applied. The

*Presented in the discussion on electric welding at a joint meeting of the American Institute of Electrical Engineers and the American Institute of Mining Engineers, held during the Midwinter Convention of the American Institute of Electrical Engineers, New York, N. Y., Feb. 19, 1919.

sleeves are made in halves and are die-formed of metal equal in thickness to, and of a length approximately equal in diameter to that of the pipe to which they are applied.

The welding was performed by the pencil-arc process, the current employed being 110-volt alternating, which was obtained primarily from the general supply system for the plant, carrying 6,600-volt three-phase current led through transformers and secondary lines to the welding reactances, furnishing the controlling medium required for the welding operation.

The electrodes used in the butt-welding operation were slag-coated and were of the same character as those extensively used in ship welding. The welding of the sleeves was performed with a low-carbon steel bare wire. The butt weld consisted of two runs of metal, the first of No. 10 and the second of No. 8 wire, the first run being carefully cleaned free of slag, first by hammering and finally by scrubbing with a wire brush until the metal showed clean and bright before applying the second run. The sleeve weld consisted of two longitudinal welds joining the two halves of the sleeve, and a weld on each end of the sleeve completely joining the sleeve to the pipe (See Fig. 2).

The application of the sleeve was determined upon for structural reasons, as the stresses likely to be encountered in an installation of this character are practically indeterminate.

One of the problems met with in handling pipe in such long lengths is the question of accurate alignment, which was accomplished in this case in the following manner: The proper level above the ground for the pipe was determined by its accessibility to the welder, and this point was marked on each leg of each pipe-carrying bent, and timber joists were securely fastened between the legs of the bents at the proper level, as in Fig. 3. The pipe lengths, approximately 20 feet long, were laid on these joists, and as the spacing of the bents along the axis of the pipe was 20 feet, this brought a joint between each pair of bents. The ends of the individual pipe lengths were then brought within about 1/10 of an inch of contact and lined up and clamped securely in position by means of a half-sleeve applied to the under side of the pipe and fastened by straps to each end of each pipe section (See Fig. 1).

This left the upper half of the butt joint ready for the first run of the weld. When one-half of the first run was completed on each joint, the clamps and the sleeve were removed and the entire 200-foot length of pipe was rolled on the joists until the other half of the butt joint was uppermost, and the first run was then completed. In consequence of the fact that it was possible to turn the pipe at will thereafter, the second run of the weld was made continuous.

It takes one welder about three hours to completely butt-weld and sleeve a 12-inch joint. The time required for other sizes would be in proportion to their diameter.

I feel that the use of the electric welding is going to be a very important feature in the development of power stations where the tendency is going to higher steam pressures, in the neighborhood of 600 pounds, as welded joints can unquestionably be made stronger than any other connection and are absolutely leakproof.—"Power."

## INSPECTING A SURRENDERED GERMAN DESTROYER "S" TYPE.

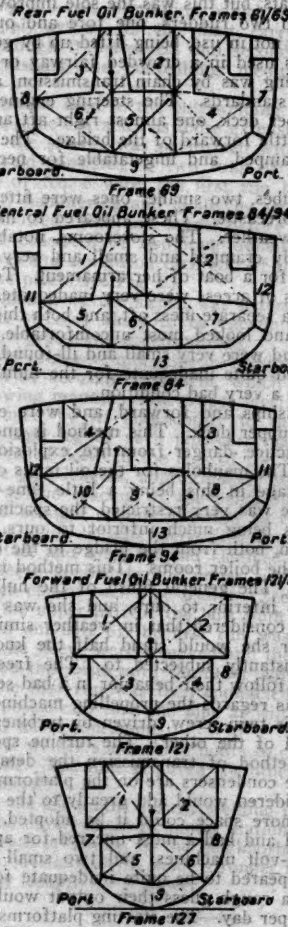
The following notes describe the impressions formed on inspecting a German "S" type destroyer built in 1914. The boat was supposed to be efficient and ready for sea in all respects, but personally the writer would

not like to take her to sea for an extended trip without an extensive refit. Considering the hull first, the hull and decks were very dirty and appeared in a very neglected condition. The bridge was certainly an improvement on the bridge of the British 1914 boats, but this was the sole improvement one could see. She was fitted with two rudders, one fore and one aft, the forward one being housed when not in use, being lifted up by gearing. Apparently the forward rudder was used in a crowded fairway or when entering and leaving harbor. Steering was by chain transmission, a very defective method according to our standards. The steering engines, two in number, were housed on the upper deck, one almost right aft and the other on the mess deck forward, a little forward of the bridge. The steering engines were vertical, much cramped, and ungetatable for necessary examination and repairs.

In addition to the main torpedo tubes, two smaller ones were fitted, one on each side at the break of the forecastle. These appeared to have no protection from heavy seas in bad weather. The storerooms, notably the engineer's storeroom, were extremely cramped and small and very dirty. The magazine seemed far too small for a boat of her armament. To both storerooms and magazines the means of access were very inadequate. The warrant officers were provided with a separate mess aft, and both this mess and the wardroom were cramped and looked most uncomfortable. The cabins opened from the wardroom and were very small and ill-found. The wardroom pantry and galleys seemed quite inadequate for the number of officers and men carried, and were in a very bad condition.

Her oil tanks were situated amidships and forward, and were carried right across the ship and up to the upper deck. This method is undoubtedly very inferior to the British practice, danger from fire, explosion and gunfire being materially increased. The manholes for the oil tanks opened to the upper deck, the only advantage in this being a little time saved when "oiling boat." The deck space was very restricted, the spacing and arrangement of the various fittings being much inferior to ours. The telegraphs were by wire transmission, both from the bridge to the engine room and from the engine room to the boiler rooms. This method is very bad and in our service is obsolete. The general finish of the hull construction throughout the vessel was inferior to ours, and she was much more lightly constructed. It is not considered that in weather similar to that in which our flotillas maneuver she would stand half the knocking about and strain our boats are constantly subjected to. The freeboard is low, and it would be interesting to follow their behavior in a bad seaway.

The following points were noted as regards the propelling machinery in the short time available. The boat is twin-screw, driven by turbines, and has two engine rooms, one forward of the other. The turbine speed is reduced to propeller speed by a method of transmission the details of which could not be ascertained. The condensers are on the platforms, not underslung, which method it is considered would add greatly to the boat's efficiency, as well as giving much more space could it be adopted. The auxiliary engines were very cramped and had a most uncared-for appearance. The dynamos were small 80-volt machines, and two small evaporators, which at a casual glance appeared to be quite inadequate for the boat's requirements, were fitted; at a rough guess their output would not be more than 24 tons of fresh water per day. The starting platforms were most restricted, and the starting wheels large and cumbersome. Engine-room telegraphs were roughly finished and, as mentioned, transmission was by wire. The ventilating and exhaust engine-room fans were of the horizontal type, and appeared of ample size to do the work required. The auxiliary feed pumps were situated in the engine rooms. The various gages seemed to be in a most inefficient condition and more were required for general efficiency. The electrical apparatus in the engine rooms




| |
|---|
| 1. Port, upper |
| 2. Central, upper |
| 3. Starboard, upper |
| 4. Port, lower |
| 5. Central, lower |
| 6. Starboard, lower |
| 7. Side tank, port |
| 8. Side tank, starboard |
| 9. Double bottom |
| 1. Port, upper |
| 2. Starboard, upper |
| 3. Port, upper |
| 4. Starboard, upper |
| 5. Port, lower |
| 6. Central, lower |
| 7. Starboard, lower |
| 8. Port, lower |
| 9. Central, lower |
| 10. Starboard lower |
| 11. Side tank, port |
| 12. Side tank, starboard |
| 13. Double bottom |
| 1. Port, upper |
| 2. Starboard, upper |
| 3. Port, lower |
| 4. Starboard, lower |
| 5. Port, lower |
| 6. Starboard, lower |
| 7. Side Tank, port |
| 8. Side tank, starboard |
| 9. Double bottom |
| Total (cubic metres) |

| Amount of Feed Water. |
|---|
| Reserve tank, after engine room |
| Waste tank, after engine room |
| Waste tank, forward engine room |
| Feed water tank I |
| Feed water tank II |
| Total (cubic metres) |

**Diagram of Oil Fuel Tanks of a German Torpedo Boat Destroyer, "S" Class.**

**A daily record is kept of amount of oil fuel in each tank, and logged in spaces at side of diagram.**

appeared up-to-date and efficient, and there seemed a good supply of electrical stores and fittings. The general condition of both engine rooms and main and auxiliary machinery was dirty in the extreme, and a total lack of care seemed to have been expended on its maintenance. Access to the engine rooms was very restricted, and several of the important auxiliaries were very inaccessible for examination and repair.

Three small-tube water-tube boilers were installed, two back-to-back in the after boiler room and the other in a separate compartment, the boilers being numbered from forward. The boiler rooms were in far worse condition even than the engine rooms and were in a deplorable state. In a long and varied experience the writer has never seen such dirty boiler rooms—a tramp steamer after a long, rough voyage would put them to shame. The chief feature of the boiler rooms is the lack of space both at the sides and at the fronts. To one used to British design it seemed impossible that efficiency could be maintained under such conditions. The sprayers for the oil fuel were much larger and more cumbersome than in our service, and the adjustment for the oil output was of a very crude nature, consisting of a lever and ball arrangement. Eight sprayers were fitted to each boiler. The forced-draft fans were of the horizontal type and when inspected with only auxiliary machinery running 1½ inches of air pressure was maintained. There appeared to be one oil-fuel pump to each boiler and one main feed pump. These latter seemed to be a bastard Weir design. Unfortunately, time did not permit of an extended examination of the boilers, uptakes and funnels. The runs of piping in both engine and boiler rooms seemed very complicated.

From a table found in the engine room the oil fuel consumption for the various speeds appeared to be as follows: 10 knots, 1.34 tons per hour; 12 knots, 1.54 tons per hour; 15 knots, 2.18 tons per hour; 20 knots, 4.55 tons per hour; 30 knots, 16.06 tons per hour. This last figure compares very unfavorably with the consumption of our 1914 destroyers. Another feature which was of interest was that there were no forced-lubrication auxiliaries, which is most extraordinary in this type of vessel. As far as one could gather, a speed of 33 knots had been obtained, but no data was available to verify this. The oil consumption and speed trials, I should imagine, had been carried out in smooth water at light draft, and with the machinery in an absolutely efficient condition. Another feature was that there were no special appliances fitted to the boilers for the purpose of making "smoke screens," and the only method they could adopt seemed to be by reducing the speed of the forced-draft fans, keeping the same quantity of liquid fuel passing through the sprayers.

It is to be hoped that at a future date more data will be available, especially as regards oil-fuel consumption, lubricating oil used, speed actually obtained and method of transmission.—"Shipbuilding and Shipping Record."

## BOOK REVIEWS.

HEAT. By E. M. SHEALY. Published by McGRAW-HILL BOOK CO., New York. 262 pp., 110 ill.

STEAM BOILERS. By E. M. SHEALY. Published by McGRAW-HILL BOOK CO., New York. 356 pp., 185 ill.

STEAM ENGINES. By E. M. SHEALY. Published by McGRAW-HILL BOOK CO., New York. 290 pp., 173 ill.

The above is a set of text books for correspondence students in the University of Wisconsin Extension Division. The set forms a very useful library for operating engineers and firemen, written in as non-technical style as possible.

"Heat" first treats the fundamental laws governing generation, transfer and transformation of heat and illustrates these laws by familiar examples in such manner as to create and hold the reader's interest. These laws are followed by a simple treatment of steam and the various media for refrigerating machinery. The principles of steam and gas engines, refrigerating machines and air compressors are treated in a practical manner adapted to the needs of the student and operator.

"Steam Boilers" is written for the practical man—the fireman, the man in charge. Fireroom equipment and the operation of boilers is stressed, little space being devoted to design. Fuels and the theory of combustion prepares the student for the sections on firing and smokeless combustion.

"Steam Engines" treats of the fundamental principles underlying the operation of the steam engine in such a manner as to enlist the interest of the average operating engineer. Of the nineteen chapters, six deal with valves and valve gear; the reason for this large proportion is given by the author because

his experience shows that this is a subject least known by operating engineers. Chapters on condensing apparatus and lubrication, engine testing and governing distinguish this book from the usual run of books in this field.

---

SHIP STABILITY AND TRIM. By PERCY A. HILLHOUSE, was reviewed in the last issue of this Journal. The U. S. publishers are D. VAN NOSTRAND Co., New York, N. Y. Price, $4.50.

---

THE MARINE STEAM TURBINE, 5th ed. By J. W. M. SOTHERN. Published by D. VAN NOSTRAND Co., New York, N. Y. 756 pp., about 700 ill.

This new edition (5th) of this well-known work has been enlarged by more than 200 pages and 350 illustrations over the 4th edition. The material of the previous edition has been thoroughly revised, and the new material incorporated includes: Impulse Turbines, Turbo-generators, Double Reduction Gearing, Marine Electric Propulsion, Michel Thrusts, Feed Regulators, Torsion Meters, recent developments in turbine practice, and a section has been added on the practical operation of marine turbines, containing much valuable data from actual tests and trials. This latter section should be of particular interest and value to the operating engineer.

Among other new material there is included a reprint of Mr. Roland S. Portham's paper, read before the Institute of Engineers and Shipbuilders in Scotland, on the "Ljungstrom turbine and its application to marine propulsion," with a full description of the installation of Ljungstrom Turbine Machinery on the S. S. *Wulsty Castle*. (See "Proc. A. S. N. E.," Nov., 1918.)

The unusual profusion of illustrations makes this book most pleasing reading and adds very materially to its value as an educational work. No pains have been spared in this respect. An excellent work for every engineer's book shelf.

RECRUIT MANUAL. By GEORGE C. THORPE, Colonel, U. S. M. C. Published by J. B. LIPPINCOTT Co., Philadelphia. 168 pp.

This manual is intended to excite the recruit's interest by answering the many questions propounded in the beginning. This is what the author terms "the psychological moment to seize upon his interest and to fix a spell upon him that will keep his interest going." General instructions to the recruit, simple methods of learning signalling, the manual of arms, the rifle and how to handle it, first aid measures, extracts from "Articles for Government of the Navy," and a military glossary are some of the features of this work. Pocket size, it is the best recruit manual we have seen.

---

MESS MANAGEMENT, by COLONEL WILLIAM E. DUNN, 344th Field Artillery. J. B. LIPPINCOTT Co. 16mo. Flexible cloth, $1.00 net.

This small volume, which can be carried in the pocket, is a guide for mess officers on how to organize and how to supervise the operation of the mess, so as to insure the men being well fed on the Government money allowance for food. It contains detailed directions on how to keep mess accounts, how to plan menus, sample menus being included, and how to plan purchases.

When it is considered that an army is said to fight on its stomach—for it is certain that no body of ill-fed troops can possibly be effective for any length of time—the importance of making the best possible use of the food-money allowance is seen to be a vital need. The secret of running a good mess is to put plenty of effective headwork into it. Colonel Dunn's practical manual will aid to make this headwork effective.

---

GEORGE WESTINGHOUSE. His Life and Achievements. By FRANCIS E. LEUPP. LITTLE, BROWN AND CO., Boston, 1918. Cloth, 6 x 9 in., 304 pp., 5 pl., 6 portraits. $3.

Mr. Westinghouse counted many readers of this Journal among his friends and this volume will hold special interest for them. It deals with the personal side of Mr. Westinghouse's career. Mr. Leupp frankly confesses that the mission of this volume is "simply human." No one could read this biography of one of the most prominent men of his time in the industrial world without receiving inspiration therefrom. It points a strong moral for the young man of today.

Compiled without the aid of diaries or files of personal correspondence, and relying upon the memory of associates, newspaper and magazine articles and the like for data, Mr. Leupp deserves great credit for giving us this true human picture of the wonderful character of this man.

Engineer, inventor, administrator, business man and father to his large industrial family, Mr. Westinghouse worked so successfully in so many lines that this work should have its companion piece dealing with the technical side of his career. Who will write it?

---

NAVAL POWER IN THE WAR. By C. C. GILL, Commander, U. S. Navy. GEORGE H. DORAN CO., New York, N. Y. 302 pp., 11 maps and diagrams, 9 ill. $1.50 net.

Commander Gill performed active service during the war in the Cruiser and Transport Service and can write as an authority on this subject. He has put the story of the Naval operations of the war to the date of signing the armistice into excellent brief shape.

The opening naval activities of the war, the Battle of Jutland and other major actions, the results of such actions, submarine and anti-submarine warfare, the part this country played in the naval activities, the transport and convoy system, all lead to the concluding section which treats of the Naval Lessons of the War.

The whole subject is graphically treated. This book is approved for use as a text book by the Academic Board of the U. S. Naval Academy.

PRACTICAL SHIP PRODUCTION. By A. W. CARMICHAEL, Lieutenant Commander, C. C., U. S. Navy. Published by McGRAW-HILL BOOK CO., New York. 250 pp., 101 ill.

This book presents in convenient form the most important general principles of ship design and construction. The treatment is practical rather than theoretical and is intended to fit the practical engineer or production man, who is well grounded in mathematics and the mechanical processes, to turn his talents to shipbuilding.

To workmen in the shipyards this volume is specially recommended. Its contents include the requirements and general descriptions of ships, structural members and design of ships, shipyards, preliminary steps in ship construction, and the building of ships. Within its limitations, quite the best book of the year in this field.

# ASSOCIATION NOTES.

## ADDRESSES.

To ensure delivery of your Journal promptly, the necessity of your keeping the Secretary advised of your address at all times cannot too strongly be urged.

The following members and associates have joined the Society since the publication of the last Journal:

## MEMBERS.

Astoria, Philip, Lieut., U. S. N.
Bandura, Oscar F., Lieut., U. S. N.
Durgin, Calvin T., Lieut., U. S. N.
Grommer, J. G. B., Lieut. Commander, U. S. N.
Hobby, Thomas F., Lieut., U. S. N.
Kershaw, Herbert E., Lieut., U. S. N.
Mayville, Armand, Lieut. (CC), U. S. N.
Owen, W. C., Commander, U. S. N.
Usina, M. N., Capt. of Engineers, C. G.

## ASSOCIATES.

Akimoff, N. W., 1013 Harrison Building, Philadelphia, Pa.
Blankenhorn, George S., 215 Drexel Building, Philadelphia, Pa.
Borum, J. V., 125 Oak St., Jacksonville, Fla.
Burch, Charles H., Lieut., U. S. N. R. F.
Chapman, Lawrence B., Electric Boat Co., Groton, Conn.
Collett, Eugene, 1134 50th St., Brooklyn, N. Y.
Cooper, Lewis M., Lieut. U. S. N. R. F.
Eriksen, O. E., 1825 Market St., Jacksonville, Fla.
Hawthorne, W. G., 115 Broadway, New York City.
Hocken, Barry, Engineer Commander, R. British Navy.

Hoyer, K., care Newport News Shipbuilding and Dry Dock Co., Newport News, Va.

Hubbard, William E., Lieut., U. S. N. R. F.

Hymans, F., Glen Ridge, N. J.

Knight, A. Raymond, South Eliot, Me.

McGarvey, Joseph T., care American and British Manufacturing Co., Bridgeport, Conn.

Petrie, G. Rollo, P. O. Box 952, Newport News, Va.

Phillips, H. M., 1211 Fern St., N. W., Washington, D. C.

Seward, Herbert L., 966 Townsend Ave., New Haven, Conn.

Slocum, Stephen E., Prof., University of Cincinnati, Cincinnati, Ohio.

Smith, Alfred O., Lieut., U. S. N. R. F.

Westwater, A., P. O. Box 237, Everett, Wash.

Yarrow, Harrold E., Yarrow & Co., Ltd., Scotstoun, Glasgow, Scotland.

## ANNOUNCEMENT.

THE TERRY STEAM TURBINE COMPANY makes the deferred announcement that since the death of Mr. Charles E. Hague in April, 1917, their interests in the Philadelphia District have been under the supervision of Mr. V. L. Sanderson, with headquarters at 605 Widener Building.

They further announce that Mr. James H. Weir has recently become associated with them as assistant to Mr. Sanderson in that district.

They earnestly solicit requests for engineering advice along all lines of turbine application.